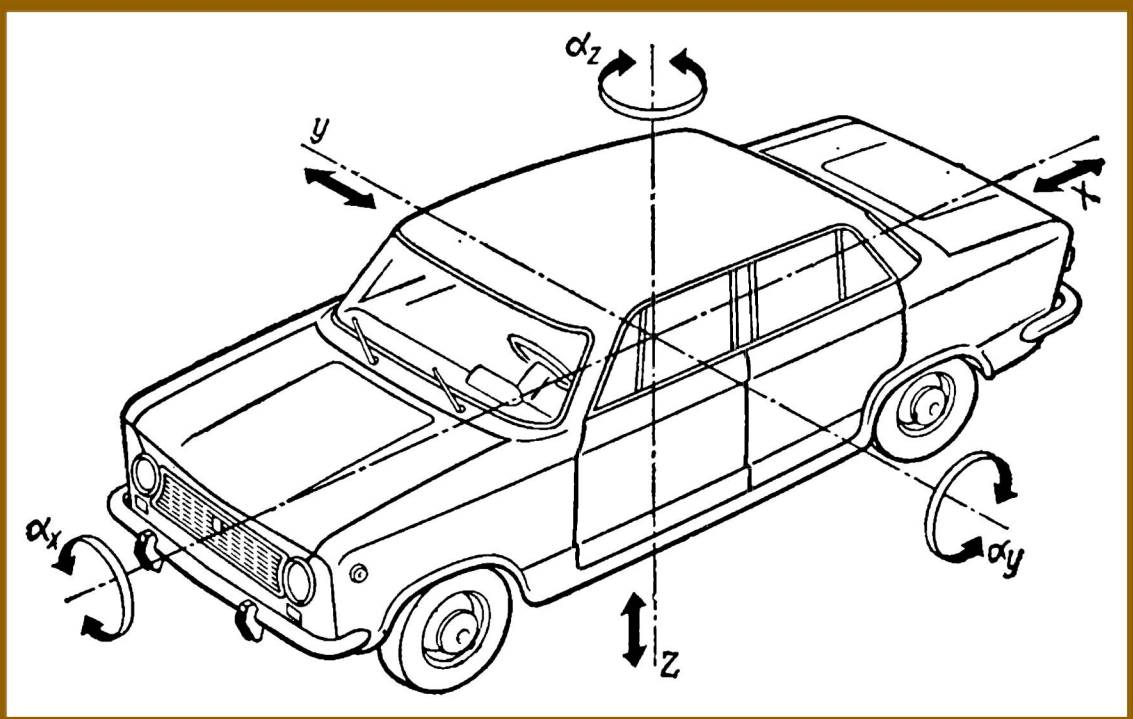
M.D. ARTAMONOV, V.A. ILARIONOV, M.M. MORIN

MOTOR VEHICLES

FUNDAMENTALS AND DESIGN



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ИЗДАТЕЛЬСТВО
МИР

М. Д. АРТАМОНОВ, В. А. ИЛАРИОНОВ, М. М. МОРИН

ОСНОВЫ ТЕОРИИ И КОНСТРУКЦИИ АВТОМОБИЛЯ

ИЗДАТЕЛЬСТВО «МАШИНОСТРОЕНИЕ» · МОСКВА

M.D. ARTAMONOV, V.A. ILARIONOV, M.M. MORIN

MOTOR VEHICLES

FUNDAMENTALS AND DESIGN

Translated from the Russian
by
A. TROITSKY

MIR PUBLISHERS · MOSCOW

First published 1976
Revised from the 1974 2nd Russian edition

The Greek Alphabet

Αα	Alpha	Ιι	Iota	Ρρ	Rho
Ββ	Beta	Κκ	Kappa	Σσ	Sigma
Γγ	Gamma	Λλ	Lambda	Ττ	Tau
Δδ	Delta	Μμ	Mu	Υυ	Upsilon
Εε	Epsilon	Νν	Nu	Φφ	Phi
Ζζ	Zeta	Ξξ	Xi	Χχ	Chi
Ηη	Eta	Οο	Omicron	Ψψ	Psi
Θθϑ	Theta	Ππ	Pi	Ωω	Omega

На английском языке

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Our address is:
USSR, 129820, Moscow, I-110, GSP, Pervy Rizhsky Pereulok, 2, Mir Publishers.

Printed in the Union of Soviet Socialist Republics

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Notation

η_t = thermal efficiency
α = excess air coefficient
μ_0 = molar change coefficient of combustible mixture
γ_r = coefficient of residual gases
η_v = admission coefficient
η_e = brake thermal efficiency
η_{tr} = direct transmission efficiency
η_{rev} = reverse transmission efficiency
φ = grip coefficient
f = coefficient of rolling resistance
ψ = coefficient of road resistance
k_{air} = air resistance coefficient (streamlining coefficient)
p_i = mean indicated pressure
p_m = mean pressure of mechanical losses
p_e = mean effective pressure
g_e = specific effective fuel consumption
ω_w = angular velocity of driving wheels
ω_e = angular velocity of crankshaft
r = radius of driving wheels
i_{tr} = gear ratio of transmission
j = acceleration of the wheel centre
α_{rd} = steepness of road ascent
i = road gradient
G_a = full weight of automobile
G = motor vehicle gravity force (weight)
D = dynamic factor
M_e = effective torque
M_f = moment of friction
M_h = moment of hydraulic losses
M_m = moment of energy losses on friction
M_t = tractive moment
M_{ax} = moment on axle shaft
M_r = moment of rolling resistance
M_a = mass of automobile
M_{in} = moment of inertia
N_e = effective power
N_m = power of mechanical losses
N_i = indicated power
N_{lit} = power-per-litre
N_t = tractive power
N_f = power expended on friction
N_{fe} = power expended on friction in engine (braking power)
N_g = power expended on gradient
N_r = power to overcome the rolling resistance
N_{rd} = power of road resistance
N_{air} = power to overcome air resistance
P_t = tractive force
P_{tr} = transmission resistance force
P_g = force of resistance due to grade
P_{gr} = force of tyre road grip
P_r = force of rolling resistance
P_{rd} = force of road resistance
P_{air} = force of air resistance
P_{in} = force of inertia
P_{tra} = resistance force of trailer
P_c = centrifugal force

Introduction

Motor vehicle science has the aim of improving the motor vehicle efficiency and lowering transportation costs. This can be achieved if the average speed of cars and trucks is increased and the fuel consumption decreased assuring also safety of motion and comfort for the driver and passengers. Knowledge of the motor vehicle fundamentals is indispensable for designing and refining new models and also for selecting appropriate types of motor vehicles. The theoretical conclusions serve as the basis for estimating travelling stock efficiency and transportation cost. These conclusions also underlie the calculation of car parts for strength and durability.

The performances show the possible effective use of a car or truck in certain conditions and allow the suitability of the design for these conditions to be assessed.

The motor vehicle fundamentals consider the performances related directly to the motion of a car or truck. Here belong the dynamic characteristics, fuel economy, steerability, stability, cross-country ability and smooth running. The other performances (holding capacity, durability, adaptability for technical maintenance and repair, etc.) are considered in other courses.

Dynamic characteristics or *speed capability* of a motor vehicle is understood as its ability to convey loads and passengers at the maximum possible average speed. The better the dynamic characteristics, the less the time required for transportation and the higher the efficiency of a car or truck.

The dynamic characteristics of a motor vehicle depend on its tractive and braking properties. They therefore form the subject of two sections, one describing the tractive and the other, the braking dynamics.

Fuel economy of a motor vehicle is its property to employ rationally the energy of the fuel. The expenses for fuel make up a considerable portion of the transportation costs. For this reason the less the fuel consumed, the cheaper the operation of a car.

Steerability is the property of a motor vehicle to change the direction of its motion as the steerable wheels are turned. Safe motion of a car or truck largely depends on the steerability.

Stability is the property of a motor vehicle to preserve the direction of motion and resist the action of forces which tend to swerve and topple it. The stability of a car as well as its steerability and braking dynamics ensures safe motion. Stability should be especially good when a motor vehicle is driven over a slippery road and at a high speed.

Cross-country ability is the property of a motor vehicle to work in arduous road conditions (snow covering, boggy land, fords) when the driving wheels do not skid or knock against road irregularities and objects. Cross-country ability is very important for motor vehicles which service agriculture, forestry, building sites and quarries.

Smooth running of a motor vehicle is its ability to move over uneven roads without intense jolting of the body. The average speed of motion and fuel consumption as well as keeping loads intact and passenger comfort appreciably depend on smooth running of the motor vehicle.

Each of these performances is as a rule considered separately, but all of them are actually closely interrelated. Any change in the design of a car or truck, undertaken for improvement, inevitably tells on the other properties. Thus, the stability of a motor vehicle can be improved if the centre of gravity is arranged as low as possible but this, however, is likely to worsen cross-country ability. The speed of a car primarily depends on its dynamic characteristics but on a slippery road it is often reduced to preserve stability and on an uneven road, to ensure smooth motion. Increase in the average speed testifies to the improved dynamic characteristics but this is often accompanied by increased fuel consumption.

The contradictory demands imposed on a car or truck impel designers to seek a compromise in improving the essential performances for any given type of a motor vehicle while worsening somewhat the properties of less importance. A motor vehicle should therefore be evaluated from the entire complex of performances and not by one property alone.

The present-day state of science and engineering makes it possible to build a motor vehicle capable of operating in any conditions. On the one hand, this is hampered by the mass scale of car production distinguished by a limited variety of produced models, and by the unification of the units mounted on various-purpose vehicles. On the other hand, each vehicle is intended to operate in diverse conditions and must possess a certain degree of universality. This explains, for example, why more and more drop-side trucks capable of carrying all kinds of load are produced today. However, to suit the national economy, the need arises for specialized motor vehicles to work in definite condi-

tions and convey loads of one type. These specialized vehicles are much more efficient than universal motor vehicles.

The operating characteristics can be determined experimentally or by calculation. To obtain data on a motor vehicle, it is tested on special stands or directly on the road in conditions close to the operating ones. These tests cost much and require qualified labour. Besides, it is extremely difficult to reproduce exactly all operating conditions. For this reason the tests of a motor vehicle are combined with a theoretical analysis of the performances and calculation of their characteristics. This analysis is especially important for the motor transport workers who handle the vehicles whose basic parameters are practically impossible to change. In order to establish the degree by which the design of a motor vehicle conforms to the operating requirements, it is necessary to determine at least approximate values of the separate characteristics. This can be accomplished more easily and faster by calculation.

A motor vehicle is a complex machine created by the efforts of many workers in various fields of science and engineering of many countries. The first motor vehicles operated by steam appeared in the 18th century and were heavy and cumbersome. In 1860, Lenoir invented the first internal-combustion engine operated by lighting gas. In 1870, Langen and Otto built four-cycle gas engines with forced ignition of the mixture, and in 1897, R. Diesel designed the first airless injection engine. In 1883, K. Benz built a car with an internal-combustion engine and in 1888 the first motorcycle designed by Daimler appeared.

Automotive engineering development was accompanied by perfection in the study of the scientific subjects related to the motor vehicle performances. In 1906, V. I. Grinevetsky devised his thermal calculation method for an engine. Later, this method was improved by E. K. Mazing, B. S. Stechkin and N. V. Inozemtsev. Much credit goes to N. R. Briling, G. G. Kalish, M. S. Khovakh, and others, for diesel engine improvements.

E. A. Chudakov, the founder of the Soviet automobile school, has written more than two hundred works on various branches of automotive science. G. V. Zimelev has studied the tractive dynamic characteristics of a car and suggested the analytical method for calculating their indices. Ya. M. Pevzner, A. S. Litvinov and Ya. E. Farobin have developed the theory of motion of a motor vehicle on turn. In the field of braking dynamics much success has been achieved by N. A. Bukharin and A. B. Gredeskul, and the problems of the smooth running have been generalized in the works of R. V. Rotenberg. N. A. Yakovlev and Ya. Kh. Zakin put in a great deal of effort to investigate the performances of a trailer train.

A close link with practice is the decisive prerequisite for the successful development of any theory. Today, G. V. Kramarenko, L. L. Afanasiev and D. P. Velikanov have been doing much work to embody the fundamentals of the theory into a concrete shape and make it applicable to the solution of the real tasks regarding the operation of motor vehicles.

The motor vehicle performances also form the subject of a close study on the part of foreign scientists.

The motor transport research institutes generalize the experience of advanced enterprises and study the most typical operating conditions to present the most scientifically substantiated demands to the models of motor vehicles newly built or modernized.

Part One
THEORY OF AUTOMOBILE ENGINES

Chapter I THEORETICAL CYCLES OF INTERNAL-COMBUSTION ENGINES

In an internal-combustion engine heat is converted into mechanical work as a result of incessant changes in the state of the working medium, i.e., gases present in its cylinders.

The combination of these changes or processes is known as the *actual cycle of an engine.*

An actual cycle consists of real processes occurring in an engine cylinder with variable values of the heat capacity of the air and products of combustion. Fuel combustion appreciably changes the chemical composition and, hence, the physical properties of the gases in the cylinder. Strictly speaking, the amount of gases during compression and expansion never remains constant because the opening and closure of the intake and exhaust valves do not as a rule coincide with the piston reaching the top dead centre (TDC) and bottom dead centre (BDC). Besides, an intensive heat exchange takes place between the gases and the cylinder walls and the piston entailing hydraulic and mechanical losses.

The complex course of real processes occurring in an engine cylinder makes their analysis difficult with usual thermodynamic formulas. For this reason *theoretical* (*thermodynamic*) *cycles* are employed to reveal and analyze the basic factors which determine the main quality indices of an engine operation.

As distinct from actual cycles, theoretical cycles are devoid of any losses except for the inevitable rejection of heat to a cool sink since, according to the second law of thermodynamics, heat cannot be converted into mechanical work without such rejection.

The following assumptions are made when considering theoretical cycles:

1. The cycles are regarded as closed, and it is assumed that a constant quantity of ideal gas is used, i.e., the processes of admission and exhaust and the resulting pumping losses are disregarded.

2. The heat capacity of gas during the entire cycle is assumed to be equal to a constant value which does not depend on temperature.

3. Combustion of fuel in an engine cylinder is replaced by an instantaneous fictitious heat input from outside and the exhaust of the waste gases, by an instantaneous fictitious heat rejection to a cool sink.

4. The processes of gas compression and expansion are assumed as adiabatic ones, i.e., without heat exchange with the external medium.

It follows from these assumptions that the efficiency of the theoretical cycles is higher than in corresponding real engines. However, the theoretical cycles are useful in elucidating the effect produced by the principal thermodynamic factors on the process of conversion of heat into mechanical work, and also in comparing various cycles from the standpoint of their effectiveness and economy.

Three theoretical cycles differing in the method of heat input to the working medium can be employed for reciprocating internal-combustion engines.

1. *Constant-volume cycle.* With this method the combustion process is close to that observed in engines with forced ignition of the mixture by an electric spark.

2. *Constant-pressure cycle.* In this case the process of combustion is close to that occurring in engines where the fuel is atomized by compressed air (it is not utilized in up-to-date engines).

3. *Constant-volume and constant-pressure cycle* (compound cycle). With this method the combustion process is analogous to the process of combustion in airless-ignition and compression-ignition engines.

According to the second law of thermodynamics, the rejection of heat to a cool sink occurs in all cycles at constant volume.

1.1. Constant-Volume Cycle

The theoretical cycle of an internal-combustion engine with heat input at constant volume shown in Fig. 1.1 consists of the following processes.

Compression of the working medium (gas) changes the pressure in the cylinder along the line *ac* without heat exchange with the environment, i.e., adiabatically. Then, section *cz* receives an external quantity of heat Q_1 at constant volume, i.e., along the

isochor, which increases the temperature and pressure of the gas. After this, the gas is adiabatically expanded along the line *zb* and heat Q_2 rejected along the line *ba* to the cool sink on the isochor. Therefore, the working medium undergoes a number of changes and returns to its initial state thus closing the cycle. The cycle may be repeated any number of times if the above conditions are observed.

The useful mechanical work of the cycle L_t is proportional to the area *aczb* restricted by the curves of the separate processes of the cycle and is equivalent to the difference between received heat Q_1 and heat Q_2 rejected to the cool sink, i.e.,

$$L_t = Q_1 - Q_2$$

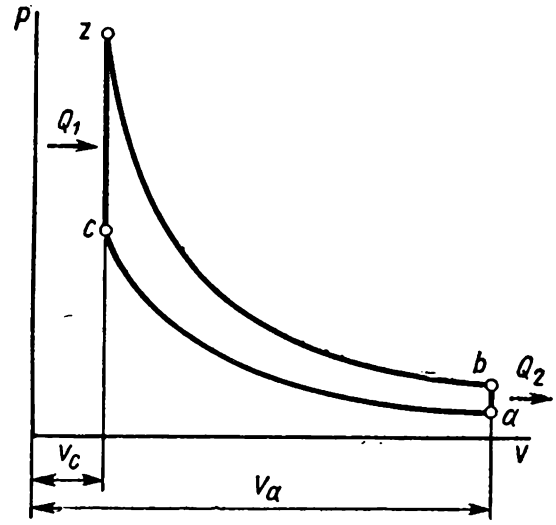

Fig. 1.1. Constant-volume cycle

The degree of utilization of received heat Q_1, as a result of its partial conversion into useful mechanical work L_t, i.e., the heat economy of the cycle, is given by the *thermal efficiency*:

$$\eta_t = \frac{L_t}{Q_1} = 1 - \frac{Q_2}{Q_1}$$

Assuming 1 kg of working medium for heats Q_1 and Q_2 we may write for the cycle being considered:

$$Q_1 = c_v (T_z - T_c)$$
$$Q_2 = c_v (T_b - T_a)$$

where c_v is the specific heat at constant volume.

Substituting the obtained expressions for Q_1 and Q_2 into the expression for η_t we get

$$\eta_t = 1 - \frac{c_v (T_b - T_a)}{c_v (T_z - T_c)} = 1 - \frac{T_b - T_a}{T_z - T_c}$$

From thermodynamic theory we know that for adiabatic compression and expansion

$$T_c = T_a \varepsilon^{k-1} \text{ and } T_z = T_b \varepsilon^{k-1}$$

where $\varepsilon = \frac{V_a}{V_c}$ = compression ratio of the engine

$k = \frac{c_p}{c_v}$ = adiabatic exponent

c_p = specific heat at constant pressure

Inserting the found values of temperatures into the formula for thermal efficiency we obtain

$$\eta_t = 1 - \frac{T_b - T_a}{\varepsilon^{k-1}(T_b - T_a)} = 1 - \frac{1}{\varepsilon^{k-1}} \qquad (1.1)$$

It follows from formula (1.1) that the thermal efficiency of the given theoretical cycle depends on the compression ratio ε and the adiabatic exponent k.

A growth in the values of ε and k increases the thermal efficiency of the cycle (Fig. 1.2), more intensely in the region of small compression ratio values.

A higher compression ratio increases the thermal efficiency

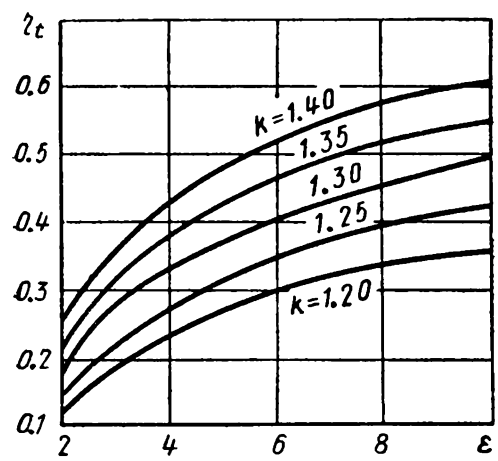

Fig. 1.2. Thermal efficiency η_t versus compression ratio ε and adiabatic exponent k forthe constant-volume cycle

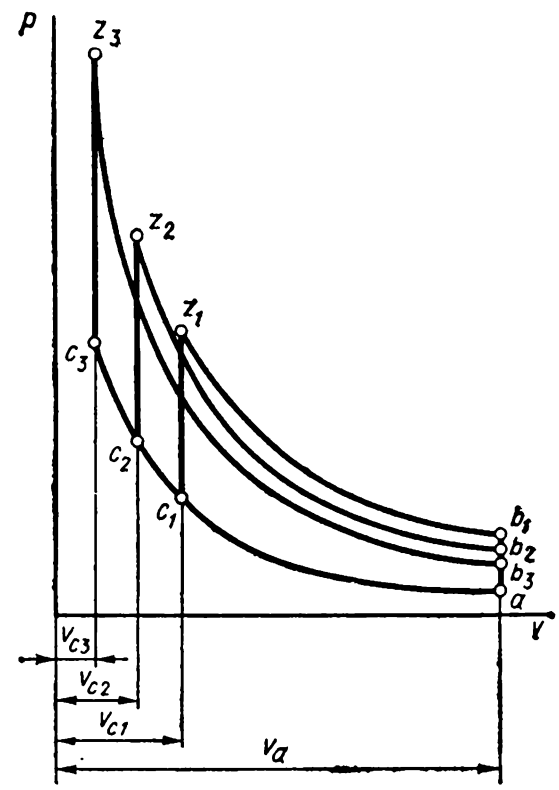

Fig. 1 3. Constant-volume cycles with various values of ε

and also the useful work of the cycle. This can be seen in Fig. 1.3, thus

$$L_t = \eta_t Q_1 \qquad (1.2)$$

The formula shows that the useful work of a cycle also depends on the quantity of received heat Q_1.

Further study of Fig. 1.3 shows that a higher value of ε reduces the gas pressure at the points b_2 and b_3 due to the drop in its temperature. In turn, this drop diminishes the quantity of heat Q_2 rejected to the cool sink, i.e., reduces the heat losses when the waste gases are exhausted.

Therefore, it is advantageous to increase the compression ratio of an engine. However, it is hardly possible in this cycle to increase the compression ratio of engines with forced mixture igni-

tion by more than 11-12 units because the higher pressure in compression and expansion considerably increases the loads and the resultant friction losses in the crank mechanism. In addition, the compression ratio of these engines is limited by the phenomena of self-ignition and detonation combustion of the mixture. For this reason the compression ratios in engines with forced mixture ignition are kept within 4-5 for kerosene carburettors, within 6-12 for petrol carburettors and 8-11 for gas engines.

1.2. Compound Cycle

In the compound cycle, heat to the working medium is partly supplied at constant volume and partly at constant pressure (Fig. 1.4). This cycle underlies the operation of compression-ignition engines (airless-injection diesels).

The working medium is compressed at the point a adiabatically along the line ac after which it receives heat Q_1 equal to the sum $Q_1' + Q_1''$.

Heat $Q_1' = c_v(T_{z'} - T_c)$ is supplied along the line cz' at constant volume, and heat $Q_1'' = c_p(T_z - T_{z'})$ along the line $z'z$ at constant pressure. Then follows adiabatic expansion along the line zb and rejection of heat $Q_2 = c_v(T_b - T_a)$ into the cool sink along the line ba.

The thermal efficiency of a compound cycle can be determined from the formulas:

$$\eta_t = 1 - \frac{Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1' + Q_1''}$$

$$\eta_t = 1 - \frac{c_v(T_b - T_a)}{c_v(T_{z'} - T_c) + c_p(T_z - T_{z'})}$$

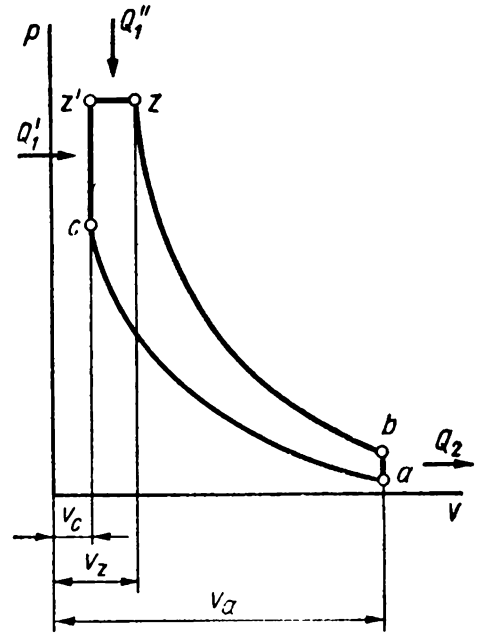

Fig. 1.4. Compound cycle

Expressing the temperature at separate points of the cycle through the temperatures T_a and T_b and after certain transformations, the thermal efficiency formula can be obtained for the compound cycle:

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}} \frac{\lambda\rho^k - 1}{\lambda - 1 + k\lambda(\rho - 1)} \tag{1.3}$$

where $\lambda = \frac{p_z}{p_c}$ = pressure increase ratio

$\rho = \frac{V_z}{V_c}$ = preliminary expansion ratio

It follows from formula (1.3) that the thermal efficiency of the compound cycle increases with a higher compression ratio ε and a growth in the pressure ratio λ, and also with a drop in the preliminary expansion ratio ρ.

The cycles with heat input at constant volume and constant pressure may be regarded as particular cases of the compound cycle.

If it is taken for granted that the volume $V_z = V_c$ and the coefficient $\rho = 1$, the thermal efficiency formula will take the form of the thermal efficiency of a cycle with heat input at constant volume:

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}}$$

If, on the other hand, it is assumed that $p_z = p_c$ and the coefficient $\lambda = 1$, this formula will describe the thermal efficiency of a cycle with heat input at constant pressure:

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}} \frac{\rho^k - 1}{k(\rho - 1)}$$

Chapter II COMBUSTION OF FUEL

Power for motor vehicles is generated from petrol for carburettor engines, from diesel fuel for diesel engines and compressed or liquefied combustible gases, or obtained by gasification of solid fuels, for gas engines.

2.1. Reaction of Burning

As is known, chemically, liquid fuel is a compound of three basic elements: carbon C, hydrogen H and oxygen O.

Denoting the mass parts of separate components by the symbols of the respective elements, the following equality may be written for 1 kg of liquid fuel:

$$C + H + O = 1$$

Depending on the amount of oxygen admitted into the cylinders of an engine, combustion of fuel may be complete or incomplete. Complete combustion yields products consisting of carbon dioxide CO_2, water vapour H_2O, surplus oxygen O_2 and nitrogen N_2 which enters the cylinder together with the air.

In the case of incomplete combustion the lack of oxygen allows some of the carbon to burn completely and form carbon dioxide CO_2 ($C + O_2 = CO_2$) and the remainder forms carbon monoxide CO ($2C + O_2 = 2CO$).

Calculating in kilomoles (kmole)*, the theoretical quantity of oxygen needed for the combustion of 1 kg of fuel equals

$$\frac{C}{12} + \frac{H}{4} - \frac{O_f}{32}$$

where $\frac{O_f}{32}$ is the number of kilomoles of oxygen contained in the fuel.

By volume, air contains 21 per cent of oxygen and 79 per cent of nitrogen and for this reason the quantity of air in kmole theoretically needed for the combustion of 1 kg of fuel

$$L_0 = \frac{1}{0.21}\left(\frac{C}{12} + \frac{H}{4} - \frac{O_f}{32}\right) \qquad (2.1)$$

* Calculation in kilomoles is tantamount to that in volume units but does not require reservations as to the temperature and pressure of gas.

The value of L_0 (in kmole/kg) is 0.512 for petrol and 0.496 for diesel fuel.

In actual fact, for each kilogramme or cubic metre of fuel the engine cylinder receives air in a quantity other than that theoretically necessary. The ratio of the actual amount of air L to the theoretically necessary L_0 is known as the *excess air coefficient*:

$$\alpha = \frac{L}{L_0}$$

If the coefficient $\alpha = 1$ complete combustion of fuel (to the final products of oxidation) is possible only when fuel and air are mixed ideally and each particle of fuel is provided with the amount of oxygen needed for its combustion. Therefore, full combustion of fuel is practically possible when the coefficient $\alpha > 1$.

To increase the power of carburettor engines and engines with fuel injection and forced ignition, a mixture is prepared with the coefficient $\alpha < 1$. In this case part of the fuel carbon burns and carbon monoxide CO is formed, as a consequence, the efficiency of the engine is impaired.

The values of the excess air coefficient α with full fuel supply vary within 0.85-1.15 for carburettor, 1.1-1.3 for gas and 1.3-1.7 for diesel engines (at small load the coefficient α is equal to 4 units and over).

2.2. Combustible Mixture and Products of Combustion

The combustible mixture in carburettor engines consists of air and fuel vapours. If the molecular mass of fuel is denoted by m_f the number of kilomoles of the combustible mixture per kilogramme of fuel can be found from the formula:

$$M_1 = \alpha L_0 + \frac{1}{m_f} \tag{2.2}$$

Since the molecular mass of petrol $m_f = 110\text{-}120$, tentative calculations can be done neglecting the value of $\frac{1}{m_f}$.

In gas engines, the combustible mixture consists of 1 kmole of gas and αL_0 kmole of air. For this reason the amount of gas-air mixture in kmole/kg will be

$$M_1 = 1 + \alpha L_0 \tag{2.3}$$

As compared with the volume of air, the volume of fuel injected into the cylinder of diesel engines is negligibly small (up to 0.01 per cent of the air volume) and the value of $1/m_f$ can therefore be

neglected. As a result, the number of kilomoles of a fresh charge can be determined by the formula:

$$M_1 = \alpha L_0 \tag{2.4}$$

The total amount of complete combustion products both of liquid and gaseous fuel can be found with the aid of the expressions considered above.

The composition of the combustion products of 1 kg of liquid fuel as expressed in kmole/kg is determined from the formulas:

$$M_{CO_2} = \frac{C}{12}$$

$$M_{H_2O} = \frac{H}{2}$$

$$M_{O_2} = 0.21(\alpha - 1) L_0$$

$$M_{N_2} = 0.79\alpha L_0$$

The total amount of complete combustion products of liquid fuel is determined as the sum of kilomoles of the separate components of the combustion products:

$$M_2 = M_{CO_2} + M_{H_2O} + M_{O_2} + M_{N_2}$$

or

$$M_2 = \frac{C}{12} + \frac{H}{2} + 0.21(\alpha - 1) L_0 + 0.79\alpha L_0$$

After transformation we obtain

$$M_2 = \frac{C}{12} + \frac{H}{2} + (\alpha - 0.21) L_0 \tag{2.5}$$

The mass of the combustion products is equal to the sum of the masses of fuel and air before combustion, but the number of kilomoles of the combustion products may not necessarily be equal to the number of kilomoles of the fresh charge.

The change in the number of kilomoles in a fresh charge due to combustion is determined as the difference:

$$\Delta M = M_2 - M_1 \tag{2.6}$$

As a result of liquid fuel combustion, the volume of the combustion products proves larger than the volume of the fresh charge, and after gaseous fuel combustion the volume (number of kilomoles) of the combustion products may be less than, equal to or larger than the volume (number of kilomoles) of the fresh charge.

The change in the volume (number of kilomoles) depends on the nature and quantity of hydrocarbons in the fuel and also on

the relationship between the amount of hydrocarbons and the amount of hydrogen and carbon monoxide.

The volume change upon combustion describes the *molar change coefficient of combustible mixture*, μ_0, which is the ratio of the number of kilomoles of the combustion products M_2 to the number of kilomoles of the combustible mixture M_1, i.e.,

$$\mu_0 = \frac{M_2}{M_1} = 1 + \frac{\Delta M}{M_1} \tag{2.7}$$

At the end of the compression stroke the cylinder contains a working mixture consisting of combustible mixture and residual gases. However, the coefficient μ_0 does not take into account the effect of the residual gases. For this reason, use is made of the *molar change coefficient of working mixture*, μ, which is the ratio of the total number of kilomoles of gases in the cylinder after combustion (combustion products M_2 and residual gases M_r left from the previous cycle) to the total number of gas kilomoles before combustion (combustible mixture M_1 and residual gases M_r):

$$\mu = \frac{M_2 + M_r}{M_1 + M_r}$$

Let us divide this expression by M_1 bearing in mind that

$$\mu_0 = \frac{M_2}{M_1} \text{ and } \gamma_r = \frac{M_r}{M_1}$$

where γ_r is the coefficient of residual gases.

Inserting the coefficients μ_0 and γ_r instead of the ratios of the quantities of gases in the formula for μ, we obtain

$$\mu = \frac{\mu_0 + \gamma_r}{1 + \gamma_r} \tag{2.8}$$

A higher μ coefficient means a greater volume of the gases generated by combustion and helps to improve the useful work of the cycle.

The value of the coefficient μ depends on the chemical composition of the fuel, the coefficient of the residual gases γ_r and the excess air coefficient α. The principal factor which affects the coefficients μ_0 and μ is the coefficient α; if it is decreased the values of μ_0 and μ grow in magnitude. For this reason, higher values of μ_0 and μ tend to increase the power of carburettor engines when they operate with a rich mixture, i.e., $\alpha < 1$.

The values of the molar change coefficient in working mixture, μ, are illustrated below:

Carburettor engines	1.02-1.12
Diesel engines	1.01-1.06
Engines operated by generator gas . .	0.92-0.98

2.3. Heat of Combustion (Heating Value) of Fuel and Mixture

The lower heating value (in kJ/kg) for liquid and solid fuels is computed from D. I. Mendeleyev's formula:

$$H_u = 34{,}013C + 125{,}600H - 10{,}900\,(O - S) - 2512\,(9H + W) \quad (2.9)$$

where C, H, O, S and W are the percentage content of separate elements and water in the fuel.

Burning the fuel in a calorimeter is the principal way of finding its heating value.

The power of engines operated by various fuels depends on the heat of combustion of the fuel mixture and the quantity of air needed for combustion.

If the volume of fuel is neglected the heating value (in kJ/m^3) of a liquid fuel-air mixture can be found from the formula:

$$H_{mix} = \frac{H_u}{\alpha L_0'} \quad (2.10)$$

where L_0' is the amount of air by volume theoretically necessary to burn 1 kg of fuel (m^3 of air/kg of fuel).

The heat of combustion of a gaseous fuel-air mixture is determined from the formula:

$$H_{mix} = \frac{H_u}{1 + \alpha L_0''} \quad (2.11)$$

where L_0'' is the amount of air by volume theoretically needed to burn 1 m^3 of gaseous fuel (m^3 of air/m^3 of gas).

2.4. Heat Capacity of Charge and Combustion Products

The temperature of combustion can be found if we know the kilomolar heat capacities of various gases which, in turn, depend on the temperature.

Distinction is made between the true and mean kilomolar heat capacities of 1 kmole of gas.

The true heat capacity is that of a gas at a definite moment with a specified temperature.

The mean heat capacity is a certain mean value of the heat capacities between two pre-assigned temperatures, assuming tentatively that the heat capacity changes according to a linear law within this temperature range.

The values of mean kilomolar heat capacities at constant volume in kJ/(kmole K) can be found from the formula:

$$mc_v = a + bT \tag{2.12}$$

where a and b = constant coefficients
T = temperature in K

The mean kilomolar heat capacities at constant pressure mc_p in kJ/(kmole K) are determined from the formula:

$$mc_p = mc_v + mR = mc_v + 8314$$

The heat capacity of a fresh charge for all engines is usually assumed to be equal to the heat capacity of air, i.e., neglecting the effect of the fuel vapours (in carburettor engines) and the difference in the heat capacities of gaseous fuel and air (in gas engines).

Chapter III ACTUAL CYCLES OF INTERNAL-COMBUSTION ENGINES

The actual cycles of engines noticeably differ from the corresponding theoretical cycles discussed above. This can be attributed to the fact that it is impossible to create in a real engine the conditions in which a theoretical cycle proceeds. For example, when considering a theoretical cycle it is assumed that the composition and quantity of an ideal gas remain constant although actually the amount of gases and their chemical composition undergo a change. After an actual cycle is terminated, the waste gases do not return to their original state and do not remain in the cylinder but are expelled into the atmosphere leaving space for a fresh charge. Therefore, an actual cycle is in essence an open cycle.

The heat capacities of gases in an actual cycle are never constant, since the temperature and the composition of gases are appreciably altered. In addition, thermal and hydraulic losses occur in an actual cycle.

Figure 3.1 illustrates the indicator diagrams of the actual cycles of a four-stroke carburettor engine (Fig. 3.1*a*) and a four-stroke diesel engine (Fig. 3.1*b*). These cycles can be represented in the form of the following alternating and partially overlapping main processes: admission (*ra*), compression (*ac'*), combustion (*c'z*), expansion (*zb'*) and exhaust (*b'r*).

The process of admission usually begins near TDC (before the point *r*) and ends near the point *M*. The process of compression ends at the point *c'* (at the moment of ignition). The process of combustion then sets in and ends on the line of expansion *zb*. The process of expansion ends at the point *b'* (the moment the exhaust valve opens) and the process of discharge usually occurs after TDC (past the point *r*).

The mechanical work performed by the gases in an engine cylinder during one cycle can be determined from the difference in the areas *bczb* and *brab*. The area *bczb* is positive and the area *brab* is negative because its magnitude is proportional to the work expended to overcome the resistances during exhaust and admission.

The separate processes of an engine actual cycle are discussed below.

3.1. Process of Admission

Admission (charging) fills a cylinder with combustible mixture or air.

The amount of combustible mixture or air received by the cylinder during admission depends on a number of factors the principal of which are: (1) hydraulic resistances of the admission and exhaust systems; (2) heating of a fresh charge by contact

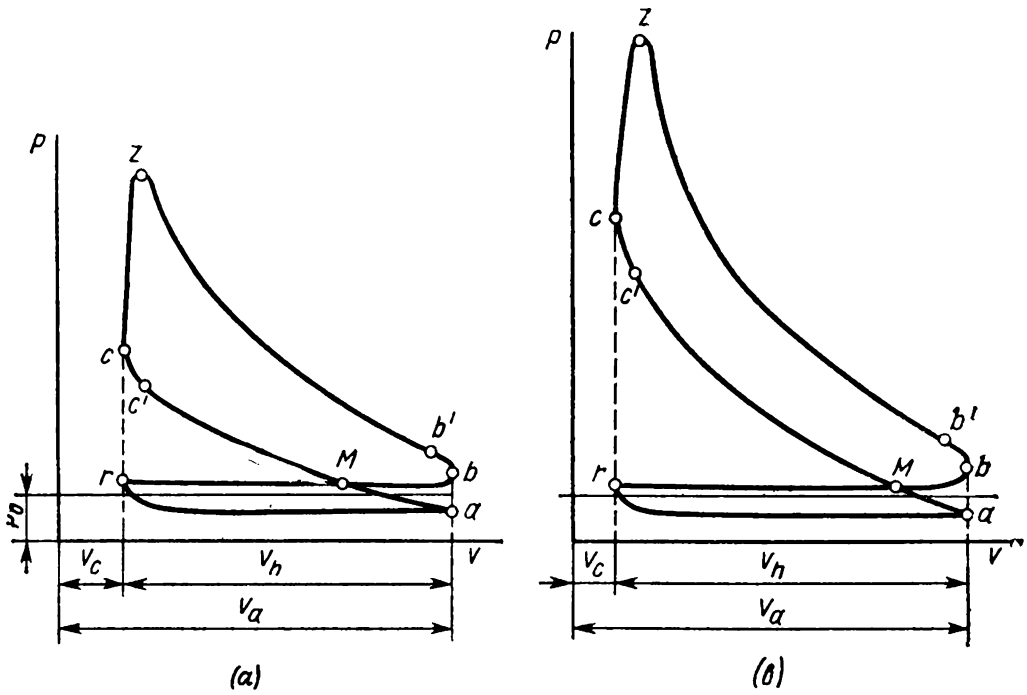

Fig. 3.1. Indicator diagrams of actual cycles for four-stroke engines
(*a*) carburettor engine; (*b*) diesel engine

with the hot parts of an engine and by its mixing with the residual waste gases left after the previous cycle; (3) presence of residual gases in the cylinder as it begins to be filled with a fresh charge.

Pressure During Admission

Before a fresh charge is admitted, the cylinder contains a certain amount of residual gases. In a four-stroke engine these gases occupy a volume of V_c in the combustion chamber (Fig. 3.2), their pressure p_r being larger than the atmospheric one and the temperature much higher than that of a fresh charge.

When the piston moves from TDC to BDC the pressure of the residual gases p_r drops to the atmospheric pressure p_0, after which the cylinder is filled with a fresh charge. In this case the pressure of the fresh charge becomes less than the atmospheric one due to the hydraulic resistances in the exhaust system.

As distinct from four-stroke engines, the process of charging in two-stroke engines occurs as a result of a forced delivery of a fresh charge into the cylinder during scavenging.

During the period of admission the pressure in the cylinder is less than the pressure of the ambient medium, the maximum rarefaction corresponding approximately to half the stroke of the piston, i.e., to the maximum piston speed. Towards the end of admission the pressure p_a slightly rises at the point a due to the velocity head built up by the forces of inertia of the charge admitted into the cylinder.

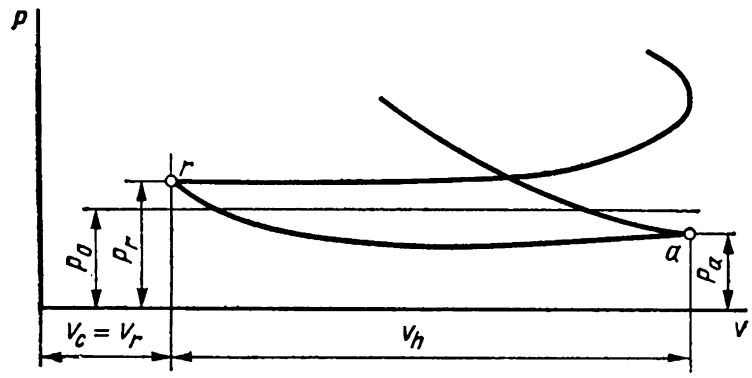

Fig. 3.2. Diagram of admission and exhaust processes occurring in the cylinder of a four-stroke engine

The drop in pressure brought about by the resistances in the intake system is

$$\Delta p_a = p_0 - p_a$$

where p_0 = atmospheric pressure in MN/m²
p_a = pressure at the point a in MN/m²

The value of Δp_a in four-stroke engines without supercharging varies within (0.10-0.25) p_a (the larger values refer to carburettor and gas engines and the smaller ones to diesel engines).

In the case of four-stroke supercharged engines, where a fresh charge enters the intake system from a supercharger under the pressure p'_a the value of Δp_a lies within $(0.05\text{-}0.10)p_a$.

Thus, the pressures at the end of admission are expressed by the formulas:

$$p_a = p_0 - \Delta p_a; \quad p_a = p'_a - \Delta p_a$$

The density and, therefore, the mass of a charge increases with higher values of the pressure p_a. For this reason, in designing an engine, everything is done to increase the admission pressure by increasing the area through the intake valve, improving the finish of the internal surfaces of parts in the intake system, decreasing the length and number of bends of the intake pipe, selecting effective valve timing phases, reducing pressure at the axhaust and by other methods.

Temperature of Gases During Admission

As the cylinder is being filled, the temperature of a fresh charge changes. As a result of fuel vaporization, the temperature of the charge drops in carburettor engines. At the same time the fresh charge admitted into the cylinder is heated when it comes in contact with the hot parts of the engine and mixes with residual gases. As a result, the temperature of the charge increases during the process of admission.

The degree of charge heating ΔT depends on the load, angular velocity of the crankshaft and cooling conditions of the engine. A growth in load is attended by an increase in the mean temperature of the cycle, which increases the degree of charge heating. When the angular velocity of the crankshaft grows, the degree of heating diminishes. Obviously, heating reduces the density of the charge and thus decreases the amount of the combustible mixture or air admitted into the engine cylinder. According to experimental data, the value of ΔT for carburettor engines lies within 10-40° and for diesel engines, within 10-20°.

A fresh charge with a temperature of $T_0 + \Delta T$ mixes in the engine cylinder with the residual gases which at a pressure of p_r are heated to temperature T_r exceeding the temperature of the charge.

The amount of residual gases at the end of exhaust is estimated by the so-called *coefficient of residual gases* γ_r which is the ratio of the quantity of the residual gases M_r to the quantity of the fresh charge M_1 admitted into the engine cylinder during charging:

$$\gamma_r = \frac{M_r}{M_1}$$

where M_r and M_1 are expressed in kilomoles.

The coefficient γ_r grows in magnitude with an increase in the volume of the combustion chamber, i.e., as the compression ratio ε drops and the pressure of the residual gases p_r increases.

If the cylinder is not scavenged and the engine operates under a full load, the coefficient of residual gases γ_r of four-stroke engines lies within the following limits:

Carburettor engines	0.06-0.12
Diesel engines	0.03-0.06

In two-stroke engines, the degree of cylinder scavenging depends on the accepted scavenging system and varies from 0.03 to 0.40.

To find the temperature T_a of the fresh charge at the end of admission, write the heat balance equation for the point a of the diagram:

$$M_1 mc'_p (T_0 + \Delta T) + M_r mc''_p T_r = (M_1 + M_r) mc_p T_a \qquad (3.1)$$

where M_1 and M_r = number of kilomoles of a fresh charge and residual gases

mc'_p, mc''_p, mc_p = mean kilomolar heat capacities of a fresh charge, residual gases and working mixture at constant pressure in kJ/(kmole K)

T_0 = temperature of the charge before it is admitted into the cylinder, K

$T_0 + \Delta T$ = temperature of the charge with regard to its heating upon contact with hot parts of engine, K

Since the quantity of residual gases is small as compared with the quantity of combustible mixture, the difference in the heat capacities may be neglected. Then, dividing all terms in equation (3.1) by M_1 we obtain

$$T_0 + \Delta T + \frac{M_r}{M_1} T_r = \left(1 + \frac{M_r}{M_1}\right) T_a$$

But since the ratio $\frac{M_r}{M_1} = \gamma_r$, then solving the latter equation with respect to the temperature of the charge T_a (K) at the end of admission we find that

$$T_a = \frac{T_0 + \Delta T + \gamma_r T_r}{1 + \gamma_r} \qquad (3.2)$$

It follows from formula (3.2) that the temperature of the charge increases at the end of admission with higher values of ΔT and γ_r. Formula (3.2) can be used to determine the temperature T_a both for four-stroke and two-stroke engines.

The values of the temperatures T_r and T_a (K) are given below:

	T_r	T_a
Carburettor engines	900-1100	340-400
Diesel engines	700-900	310-360

Coefficient of Admission (Charge Efficiency)

As a result of heating and change in the density of a fresh charge, its quantity actually received by the cylinder during admission is always less than the amount which could have filled the working volume of the cylinder at the pressure and temperature of the ambient medium

The degree of filling the cylinder with a fresh charge is estimated by the admission coefficient η_V which is the ratio of the quantity of a fresh charge actually entering the cylinder to the theoretically possible quantity which could have filled the working volume V_h at the pressure and temperature before the intake members:

$$\eta_V = \frac{G_1}{G_0} = \frac{M_1}{M_0}$$

where G_1 and M_1 = quantity of a fresh charge actually admitted into the cylinder in kg or kmole

G_0 and M_0 = quantity of a fresh charge which could have filled the working volume V_h at the pressure and temperature before the intake members in kg or kmole

According to the definition of the coefficient η_V, a fresh charge at a temperature of T_0 and a pressure of p_0 occupies a volume of $\eta_V V_h$.

On the basis of the characteristic equation $pV = 8314\,MT$ it is possible to find the number of kilomoles of the fresh charge M_1, residual gases M_r and mixture M_a from the formulas:

$$M_1 = \eta_V \frac{p_0 V_h}{8314 T_0}$$

$$M_r = \frac{p_r V_r}{8314 T_r}, \qquad M_a = M_1 + M_r = \frac{p_a V_a}{8314 T_a}$$

Substituting the obtained values of M_1, M_r and M_a into equation (3.1) and neglecting the difference in the heat capacities, we get

$$\eta_V \frac{p_0 V_h (T_0 + \Delta T)}{8314 T_0} + \frac{p_r V_r T_r}{8314 T_r} = \frac{p_a V_a T_a}{8314 T_a}$$

where the pressures are expressed in N/m^2, volume V in m^3 and temperatures in K.

Bearing in mind that $V_h = (\varepsilon - 1)\,V_c$ and $V_r = V_c$ and dividing the last equation termwise by V_c, we obtain after cancellation

$$\eta_V \frac{p_0 (\varepsilon - 1)(T_0 + \Delta T)}{T_0} + p_r = \varepsilon p_a$$

This equation is used to find the admission coefficient for the cylinder of a four-stroke engine:

$$\eta_V = \frac{\varepsilon p_a - p_r}{(\varepsilon - 1) p_0} \cdot \frac{T_0}{T_0 + \Delta T} = \frac{1}{\varepsilon - 1} \left(\varepsilon \frac{p_a}{p_0} - \frac{p_r}{p_0} \right) \frac{T_0}{T_0 + \Delta T} \qquad (3.3)$$

In deriving the formula for determining the admission coefficient for the cylinder of a two-stroke engine, account should be taken of the fact that the scavenging and exhaust ports are positioned over part of the piston stroke.

It follows from equation (3.3) that the coefficient η_V depends on the pressures at the end of admission p_a and at the end of exhaust p_r, heating of the charge ΔT and the compression ratio ε.

The pressure p_a at the end of admission is the basic parameter which affects the coefficient η_V. For example, a change in pressure p_a by 0.01 MN/m^2 brings about, all other conditions being equal, a change in the coefficient η_V by 15-18 per cent, and a change in pressure p_r by the same magnitude, only by 1-2 per cent.

The value of p_a depends on the hydraulic resistance of the intake system, i.e., on the drop in pressure Δp proportional to the square of the velocity of the combustible mixture. A rise in the heating temperature ΔT of the fresh charge tends to reduce the coefficient η_V. Engine tests show that the coefficient η_V hardly depends on the compression ratio.

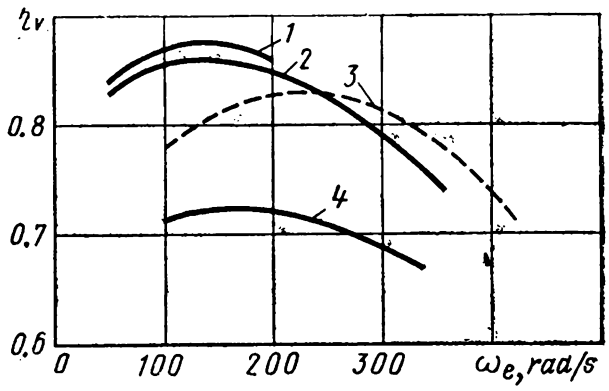

Fig. 3.3. Admission coefficient η_V versus angular velocity ω_e of engine crankshaft

1—ЯМЗ-236; *2*—ЗИЛ-130; *3*—ГАЗ-21; *4*—ГАЗ-51

The admission coefficient is also affected by the cylinder dimensions, position of the valves, design of the intakes and valve timing. If the diameter of the cylinder is large, it is possible to increase the diameter of the intake valve and reduce thereby the velocity of the charge in the intake system as well as the hydraulic losses in it.

The diameter of the cylinder can be increased by using a short-stroke engine for which the ratio of the piston stroke to the cylinder diameter is less than unity ($S/D = 0.8$-0.9). The piston speed in such engines is also less than in ordinary engines.

When overhead valves are installed, the number of the intake pipe bends may be reduced and a gradual admission of the charge into the cylinder obtained which helps to decrease hydraulic resistances in the intake system.

Figure 3.3 shows by way of example the dependences of the admission coefficient on the angular velocity ω_e of the crankshaft for carburettor engines ЗИЛ-130, ГАЗ-21 and ГАЗ-51 and also for diesel engine ЯМЗ-236.

Valve timing phases, commonly selected experimentally, assure the maximum cylinder charge within a certain angular velocity interval of the engine crankshaft. For automobile engines operating within a wide angular velocity interval the valve timing phases are chosen to suit the operating conditions of the engine.

The values of the admission coefficient η_V with an angular velocity corresponding to the maximum power of the engine are illustrated below:

Carburettor engines:	
with side valves	0.70-0.75
with overhead valves	0.70-0.85
Diesel engines	0.75-0.90

3.2. Process of Compression

The compression process creates conditions suitable for ignition and combustion of the working mixture and increases the temperature difference and the expansion ratio of the combustion products in the cycle. This favourably affects engine efficiency.

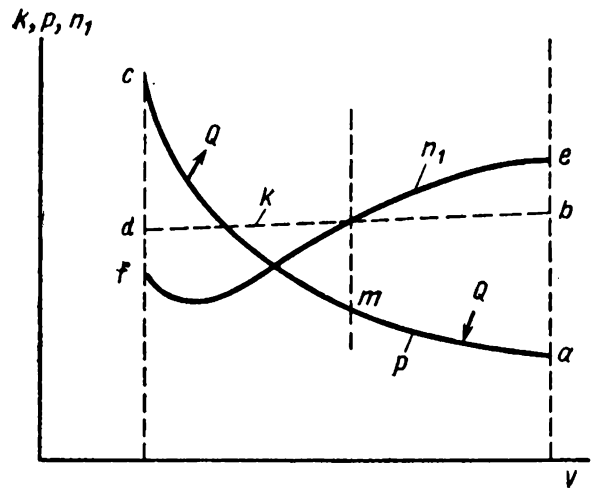

Fig. 3.4. Mutual position of compression curves, adiabatic k and polytropic n_1 compression exponents

In an actual cycle, the compression process occurs in conditions of heat exchange between the fresh charge and the parts of the engine, i.e., adiabatically. The complex nature of this heat exchange cannot be accurately expressed by thermodynamic formulas. For this reason, it is assumed that the process of compression takes a polytropic course with the exponent n_1 which remains a mean value during the entire process of compression.

Figure 3.4 shows the arbitrary mutual position of the compression curves (ac), adiabatic exponent k (bd) and the actual compression polytropic exponent n_1 (ef).

At the beginning of compression, the temperature of the fresh charge is much less than the temperature of the surrounding parts and, therefore, the first period of the compression process (section am) is accompanied by an inflow of heat from the hot parts to the charge. In this case, the polytropic exponent is larger than the adiabatic exponent ($n_1 > k$).

Upon further compression, the temperature of the charge grows and exceeds the temperature of the surrounding parts, the heat is transferred from the charge to the parts (section mc). In this case the polytropic exponent is less than the adiabatic exponent ($n_1 < k$).

Depending on the values of local temperatures, both phenomena can occur simultaneously: transfer of heat to the charge

from the more heated parts and rejection of heat from the charge to the colder ones.

It has been established from experiment that the amount of heat transferred by the charge to the surrounding parts in the section *mc* is larger than the inflow of heat from the parts to the charge in the section *am*. For this reason, the mean polytropic exponent of compression n_1 is always less than the adiabatic exponent k (the curve of the polytropic exponent of compression is under the curve of the adiabatic exponent).

The mean values of polytropic exponent of compression n_1 are illustrated below:

Carburettor engines	1.32-1.39
Gas engines	1.30-1.38
Diesel engines	1.36-1.40

The mean value of the polytropic exponent of compression n_1 depends on the angular velocity of the engine crankshaft, the size of the cylinder, the intensity of cooling, the shape of the combustion chamber and the design features of the engine. At a higher engine crankshaft angular velocity the exponent n_1 increases and, conversely, diminishes as the mean temperature of the compression process increases and the intensity of engine cooling grows in magnitude. The effect of the load and the engine compression ratio on the exponent n_1 is negligible.

The compression ratio of carburettor engines should be such that the temperature of the working mixture at the end of compression does not exceed the fuel self-ignition temperature. At the end of air compression in diesel engines the temperature should be 200-300° higher than that required for fuel self-ignition, this ensures reliable starting and steady diesel engine operation under any conditions.

The compression ratios ε are given below:

Carburettor engines	6-12
Diesel engines	16-20

The pressure at the end of compression (at the point *c*) can be found from the equation for the polytropic exponent $pV^{n_1} = \text{const}$.

The following equality may be written for the polytropic exponent of compression:

$$p_a V_a^{n_1} = p_c V_c^{n_1}$$

whence

$$p_c = p_a \left(\frac{V_a}{V_c}\right)^{n_1}$$

or

$$p_c = \varepsilon^{n_1} p_a \qquad (3.4)$$

Given below are the pressure values at the end of compression (MN/m^2):

Carburettor engines	0.8-1.5
Diesel engines	3.0-4.5

The temperature at the end of compression can be found from the equation of the polytropic process $TV^{n-1} = \text{const}$.

The following equality can be written for the polytropic exponent of compression:

$$T_a V_a^{n_1-1} = T_c V_c^{n_1-1}$$

hence

$$T_c = \varepsilon^{n_1-1} T_a \tag{3.5}$$

The temperature in carburettor engines at the end of compression varies within 600-750 K and in diesel engines, within 750-900 K.

The pressure and temperature at the end of compression depend on the compression ratio, the angular velocity of the engine crankshaft, the number of piston rings and the degree of their wear.

3.3. Process of Combustion

Combustion of a fresh charge and the subsequent expansion of the combustion products are the principal processes of an internal-combustion engine cycle. During these processes the chemical energy of the fuel is converted into heat, and part of the heat, into mechanical work. The nature of the combustion process affects appreciably the power and economic characteristics, and also wear of the engine basic elements. In turn, the course of the combustion process depends on a number of factors and primarily on the methods of mixture formation and fuel ignition; because of this the combustion process of carburettor engines and diesel engines is considered separately.

Combustion in Carburettor Engines

In internal-combustion engines with forced ignition of the mixture by an electric spark, the process of combustion proceeds not at constant volume, as in the theoretical cycle, but begins before the piston reaches TDC and terminates after it passes TDC. Thus, the process of combustion occurs near TDC and it will be more convenient to study the pressure change in this process by examining a developed indicator diagram (Fig. 3.5) which shows the pressure change in the engine cylinder relative to the crankshaft angle φ or by time. The dash line (section *3-6-7*) denotes the pressure change in the cylinder with the ignition switched off.

The asymmetry in the curves of compression *1-6* and expansion *6-7* is brought about by the heat exchange between the gases and the cylinder walls and also by partial leakage of the gases through the gaps between the piston rings and the cylinder walls.

The process of combustion begins the moment a spark jumps between the electrodes of the ignition plug marked on the diagram by point *2*, i.e., with an advance angle equal to θ.

After the spark jump the pressure in the cylinder remains for some time the same as with switched-off ignition (section *2-3*). This happens because the amount of the mixture burned in this section is negligible and the heat is transferred from the combustion products to the fresh mixture, the walls of the combustion

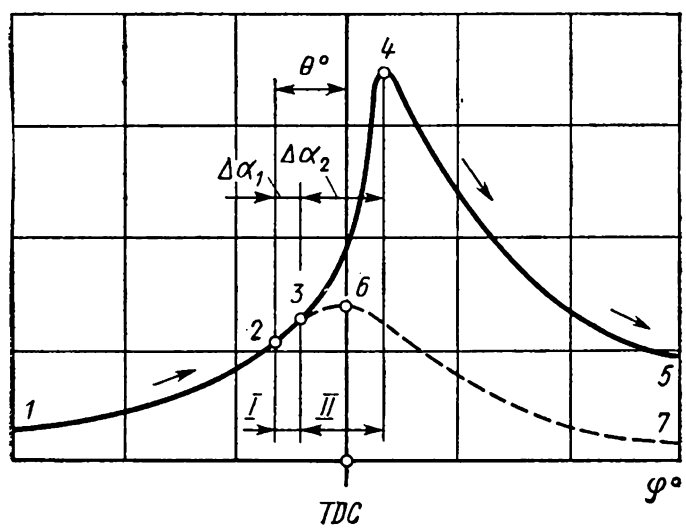

Fig. 3.5. Developed indicator diagram of a four-stroke carburettor engine

chamber and the piston crown. Then, the pressure in the combustion chamber sharply increases (section *3-4*).

The flame front does not cease to propagate in the combustion chamber at the moment when the pressure reaches its maximum value, because the mixture burns out and the heat is liberated in some places. However, this afterburning is attended by a pressure drop due to the increased cylinder volume as the piston moves down. The evolution of heat due to the afterburning of the mixture in the cylinder is usually regarded as a phenomenon related to the expansion process.

Three characteristic phases of the combustion process in an engine cylinder can be seen on the developed indicator diagram.

The first phase *I* is known as the *initial phase of combustion*. In the course of this phase preflame oxidation of the fuel occurs with a slight increase in temperature without a pressure rise. The rate of combustion during this phase mainly depends on chemical factors (properties of the fuel and composition of the mixture).

The first phase shown on the developed indicator diagram by the angle $\Delta\alpha_1$ and points *2* and *3* represents the pressure increase delay period. The duration of this phase is influenced by the excess air and residual gas coefficients, the structure of fuel molecules, energy of the ignition source, compression ratio and engine load. During the first phase of combustion about 6-8 per cent of the mixture present in the combustion chamber is burned. The duration of the first phase comprises 4-6° of the crankshaft degree.

The second phase *II* is called the *period of visible or effective combustion.* During this phase about 90 per cent of the mixture is burned and the combustion rate and pressure sharply increase in the cylinder. This phase is described by the angle $\Delta\alpha_2$. It commences at the moment when the pressure begins to increase (point *3*) and ends at the moment when pressure in the cylinder reaches its maximum value (point *4*). The duration of this phase corresponds to 20-30° of the crankshaft degree and depends on the composition of the mixture, compression ratio, ignition advance angle, shape of the combustion chamber, the mixture turbulence ratio and the engine load.

From the indicator diagram the second phase of combustion can be evaluated by the *rate of pressure rise*, i.e., the growth in pressure as referred to one degree of the crankshaft turn. The mean value of this magnitude is commonly called the *stiffness of the combustion process* determined as the ratio $\frac{\Delta p}{\Delta\alpha_2}$, where $\Delta p = p_4 - p_3$ is the difference between the maximum pressure of the cycle (point *4*) and the pressure at the beginning of visible combustion (point *3*), and $\Delta\alpha_2$ is the crankshaft rotating angle corresponding to the time elapsed from the moment when visible combustion begins to the moment when the pressure of the cycle reaches its maximum value. When the process of combustion in the cylinder of a carburettor engine takes a normal course, the mean value of the rate of pressure rise lies within 0.12-0.26 MN/(m²°). This magnitude depends on the compression ratio, mixture turbulence ratio, timing angle and other factors.

If the ratio $\frac{\Delta p}{\Delta\alpha_2} < 0.1$ combustion may also continue during expansion, this deteriorates the engine fuel economy. If the ratio $\frac{\Delta p}{\Delta\alpha_2} > 0.26$ the operation of the engine becomes "stiff", i.e., the cylinder pressure increases too sharply. This increases the loads on the crank mechanism, causes excessive wear and sometimes failure of its parts.

It has been found experimentally that the maximum useful area on the indicator diagram and, therefore, the maximum engine

power is obtained when the maximum cycle pressure corresponds to an angle equal to 10-15° of the crankshaft degree after TDC.

The third phase is known as *afterburning of the mixture* which occurs during the expansion along the line *4-5*. The duration of the third phase is short in carburettor engines and it is extremely difficult to determine its termination on the indicator diagram because the moment of complete combustion of the mixture must be known for this purpose. The third phase duration depends on the mixture composition, ignition advance angle and mixture turbulence ratio.

The mixture combustion time (in seconds) can be found from the formula:

$$\tau_c = \frac{\pi \Delta\alpha}{180\omega_e}$$

where $\Delta\alpha$ = crankshaft degree

ω_e = crankshaft angular velocity in rad/s

For carburettor engines the mixture combustion time τ_c is 0.001-0.002 s.

The mixture combustion time in an engine cylinder depends primarily on the propagation velocity of the flame front (rate of combustion). During the various periods of the combustion process the rate of combustion never remains constant but varies rather widely. On the average, when the combustion process in a carburettor engine proceeds normally, its values are within 20-30 m/s. More often than not the mean rate of combustion is tentatively determined as the ratio of the flame path, as the flame passes from the spark plug to the farthest point in the combustion chamber, to the time elapsed from ignition up to the moment when the pressure of the cycle reaches its maximum value.

Factors Affecting the Process of Combustion

The rate of combustion depends on the composition and turbulence ratio of the working mixture, compression ratio, shape of the combustion chamber, timing angle, position of the spark plug, and engine load.

The composition of the working mixture described by the excess air and residual gas coefficients has a noticeable effect on the combustion process and the flame front propagation rate.

Definite relationships exist between the quantities of fuel and air in the mixture at which the combustible mixture can be ignited in the cylinders of a carburettor engine. These boundaries are known as the *upper and lower limits of combustibility.*

These limits are not constant for various fuels and depend on the temperature, pressure and turbulence ratio of the mixture. For example, the upper limit for petrol is restricted in normal

conditions by the excess air coefficients equal to 0.3-0.5 and the lower limit by those equal to 1.2-1.3.

When the temperature of the combustible mixture increases the limits of combustibility get slightly wider, and when pressure is increased — narrower. The residual gases in the combustible mixture narrow down the limits of combustibility.

Figure 3.6*a* and *b* illustrates experimental graphs of the change in the combustion rate of petrol-air mixture in an engine and in a bomb depending on the excess air coefficient α.

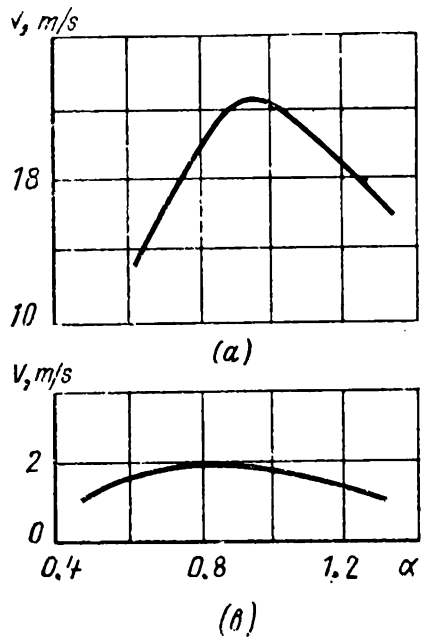

Fig. 3.6. Change in the combustion rate of petrol-air mixture versus excess air coefficient α
(*a*) in an engine cylinder; (*b*) in a bomb

The graphs show that the combustion rate is maximum if the excess air coefficients range from 0.8 to 0.9. The propagation velocity of the flame front in an engine cylinder with mixture turbulence is 8-12 times larger than in a bomb where vortex motion is absent.

If the values of the coefficient α are within 0.8-0.9, the time elapsed before the pressure begins to increase sharply proves to be minimum and the visible combustion is observed near TDC if the timing angle is selected correctly. This case ensures the maximum pressure of combustion and, as a result, the engine develops the highest power. However, the fuel economy is somewhat impaired.

When the coefficient α deviates from these values making the mixture rich or weak, the mixture combustion rate reduces. This can mainly be explained by the drop in the reaction temperature: when the engine operates with rich mixture, due to chemically incomplete combustion, and with weak mixture, due to the additional expenditure of heat in raising the temperature of the excess air.

The best fuel economy of a carburettor engine is ensured by mixtures with the excess air coefficient lying between 1.05-1.15.

A rise in the initial mixture temperature is accompanied by an increase in the combustion rate due to higher rates of chemical reactions.

A growth in the initial pressure causes a slight increase in the rate of mixture combustion. If the mixture is very rich, the reverse dependence is observed, i.e., this velocity will noticeably diminish as the initial pressure grows.

The dependence of the mixture combustion rate on the compression ratio can be explained by the combined effect of the initial

temperature, initial pressure and the presence of residual gases. A higher compression ratio tends to increase the temperature and pressure at the end of compression and diminish the relative content of residual gases in the working mixture, which increases the combustion rate.

The shape of the combustion chamber also affects the combustion rate, because it governs heat rejection to the cooling medium and indicates the path passed by the flame. The less the ratio of the surface of the combustion chamber to its volume, the less the heat lost due to heat rejection to the walls of the chamber, which results in a more intensive liberation of heat and increases the rate of combustion.

The timing angle of the working mixture also exerts a great effect on the course of the combustion process. In order to terminate combustion near TDC and thus obtain the maximum expansion of gases after they have received heat, the mixture should be ignited in advance of the piston reaching TDC, i.e., at the end of compression.

If the timing angle is set correctly, visible combustion ends at the 10-15th degree of the crankshaft angle after TDC. In this case, combustion occurs at minimum volume and the useful work of the cycle and its efficiency display their maximum values.

An increase in the crankshaft angular velocity of an engine tends to increase the combustion rate of the mixture due to the more intensive turbulence. At the same time, the higher angular velocity reduces the time required for combustion. It has been established experimentally that as the crankshaft angular velocity increases, despite the rise in the combustion rate, the timing angle should be made larger.

If the working mixture to be burned is acted upon by high temperatures and pressures the normal combustion in an engine cylinder may under certain conditions pass into an explosion combustion or detonation.

Detonation is usually attended by strong metallic knocks, overheating of the engine, claps of black smoke coming from the silencer and an unsteady engine operation. An engine must never work under such conditions as it intensifies wear and damages the parts of the crank mechanism. Prolonged operation at detonating combustion reduces power and increases fuel consumption, causes burning of the pistons and pitting of the antifriction lining of the crankshaft bearing shells.

Detonation is caused by the formation of active peroxides which are very unstable compounds, i.e., primary products of oxidation of hydrocarbon molecules formed as a result of interaction of the

active molecules of oxygen and fuel. The mixture combustion rate in detonation reaches the detonation wave velocity within 1500-2000 m/s.

Combustion in Diesel Engines

The process of ignition and combustion of fuel in diesel engines appreciably differs from that observed in engines with external mixture formation and occurs in more complicated conditions, since the cylinder contains air and not a previously prepared working mixture.

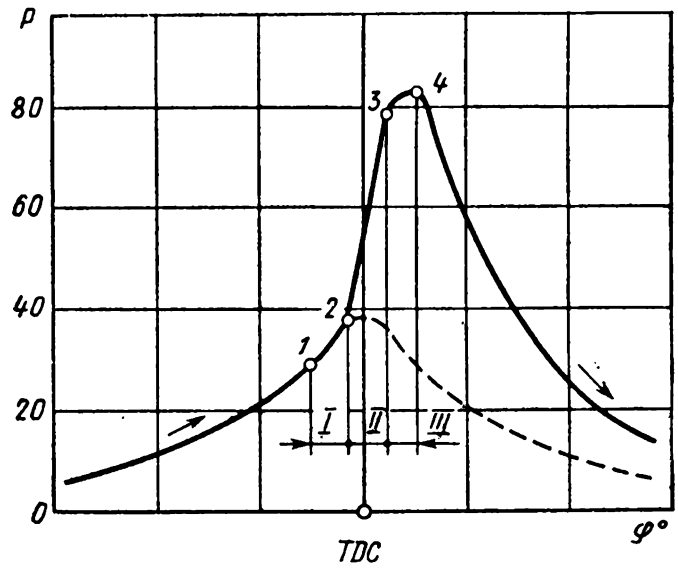

Fig. 3.7. Developed indicator diagram of a four-stroke diesel engine

The process of combustion can be subdivided into four phases (Fig. 3.7) during which the fuel is prepared for combustion, its main portion is burned and the remaining combustible components are burned out.

The first phase *I* is known as the *ignition time delay*. This phase occupies the period of time from the moment of fuel injection into the cylinder (from point *1*) to the moment of a sharp pressure increase (up to point *2*). During the first phase physical and chemical processes of preparing the fuel for combustion occur practically without any increase in pressure when compared with the compression line (*1-2*).

The duration of the first phase is affected by the physical and chemical properties of the fuel, the temperature and pressure of the compressed air and its turbulence ratio in the cylinder.

The second phase *II* is characterized by its *intensive liberation of heat and sharp increase in pressure* (section *2-3*). The intensive heat liberation takes place because of combustion of the pre-

pared fuel admitted into the cylinder during the ignition time delay and also due to partial combustion of the incoming fuel. Injection of fuel into the cylinder may continue during the entire second phase or cease before this phase ends. The rate of pressure rise during the second phase depends not only on the ignition time delay but also on the delivery rate, quality of atomization and the amount of injected fuel.

The mean value of the rate of pressure rise on the combustion line *2-3*, as in an engine with forced ignition, is equal to the ratio $\frac{\Delta p}{\Delta \alpha}$. It has been found experimentally that the mean value of the ratio $\frac{\Delta p}{\Delta \alpha}$ in the second phase, at which the diesel engine operates normally, should not exceed 0.3-0.5 MN/m² per degree of crankshaft rotation. Stiff operation is accompanied by high combustion pressures and knocking; it cannot be allowed as it intensifies wear and causes breakage of the crank mechanism parts.

The best fuel economy of a diesel engine can be ensured if the cycle pressure reaches its maximum value when the crankshaft turns through 6-10° after TDC.

The course of the second phase of combustion is influenced by the ignition time delay, the fuel distribution in the combustion chamber, the amount of fuel supplied during the first phase and also by the delivery rate and the quantity of fuel injected during the second phase.

The third phase *III* corresponds to the *period of a minor continuous pressure change* (section *3-4*). It ends at the crankshaft angle when the gas temperature reaches its maximum value. Injection of fuel is usually terminated before the beginning of the third phase.

During the third phase the combustion process has a reduced heat evolution rate as compared with the second phase, since the oxygen concentration diminishes and the combustible mixture mixes with the combustion products. The temperature of the gas, however, continually increases in this phase.

The duration of the third phase depends on the characteristic of the fuel, the turbulence ratio of the charge and the excess air coefficient.

If optimum conditions are ensured for the fuel delivery and the combustion process the temperature at the end of combustion reaches its maximum when the crankshaft turns through 20-35° after TDC.

The fourth phase *IV* is the *afterburning time* and commences when the temperature of the cycle reaches its maximum. Afterburning occurs in all diesel engines, but it is longer in high-

speed engines than in low-speed ones. The end of the fourth phase coincides with the beginning of the expansion process.

Intensive afterburning of the fuel during expansion increases the temperature of the exhaust gases and means that more heat is being rejected to the cooling medium, this worsens the fuel economy of the diesel engine. In the fourth phase heat losses can be reduced if the combustion process is made more active by intensifying the turbulence of a fresh charge. A more intensive air turbulence improves mixture formation in the entire volume of the combustion chamber. This ensures an efficient combustion process. However, vortex motion alone cannot fully provide for a good quality mixture formation and the excess air coefficient α has to be increased. For four-stroke diesel engines the coefficient α lies within 1.3-1.7.

The fuel injection duration exerts a noticeable effect on the combustion process and the cycle characteristics. With a constant cyclic fuel delivery and equal crankshaft degrees, a short injection ensures the liberation of a large amount of heat during combustion and a higher rate of pressure rise than a prolonged injection.

In the case of a large timing angle the fuel is delivered at a comparatively low temperature and pressure. This increases the ignition time delay. Towards the beginning of the second phase the cylinder accumulates much fuel and the process of combustion occurs with increased values of the maximum cycle pressure, i.e., is excessively stiff. When the timing angle is small, most of the fuel is burned during expansion, this overheats the engine, increases the temperature of the exhaust gases, reduces the rate of pressure rise during combustion, impairs the power and economy characteristics of the engine.

An increase in a diesel engine load tends to reduce the excess air coefficient, since the cyclic delivery of the fuel is increased and the amount of air admitted into the cylinder practically remains constant or even is slightly reduced because it expands upon heating.

A reduction in load increases the excess air coefficient and, as distinct from the operation of a carburettor engine with a partially opened throttle, improves the fuel economy of the diesel engine.

The economic characteristics of a diesel engine can be improved if the fuel injection advance angle is increased with an increase in the crankshaft angular velocity.

Temperature and Pressure at the End of Combustion

According to the first law of thermodynamics, the heat liberated during the combustion process is expended to increase the in-

ternal energy of the gas and perform mechanical work, it also is partly withdrawn by the cooling medium.

The total amount of heat evolved in an engine cylinder is less than the heat of combustion of the burned fuel because combustion is accompanied by inevitable losses due to chemically incomplete combustion and the dissociation (splitting) of the combustion products.

To simplify calculations, it is usually assumed that the pressure change during combustion is the same as for a theoretical cycle, i.e., a compound cycle with constant volume and pressure, and for a rapid combustion cycle with constant volume.

The temperature at the end of combustion is determined by the so-called *equation of combustion* written for 1 kg of fuel introduced into the engine and representing the heat balance in the combustion section *cz*. The principle of compiling the equation of combustion is one and the same both for the compound cycle and the cycle with constant volume but the equations of combustion obtained are different.

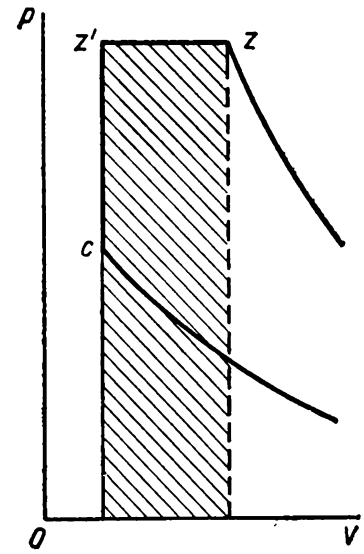

Fig. 3.8. Combustion section of an unrounded indicator diagram of a compound cycle

According to the first law of thermodynamics, the equation of combustion for a compound cycle (section *cz'z* in Fig. 3.8) can, in a general form, be written as

$$\xi H_u = U_z - U_c + L_{z'z} \tag{3.6}$$

where ξ = coefficient of active heat liberation representing the share of heat expended in the section *cz*, which increases the internal energy of gases and is converted into mechanical work

U_z and U_c = internal energy of the combustion products at the point *z* and of the working mixture at the point *c*, respectively, kJ/kg

$L_{z'z}$ = work of the gases in the section *z'z*, kJ/kg

The internal energy of the combustion products (at the point *z*) and the working mixture (at the point *c*) can be found from the formulas:

$$U_z = mc''_v (M_2 + M_r) T_z$$

$$U_c = mc'_v (M_1 + M_r) T_c$$

where mc_v'' and mc_v' are the mean kilomolar heat capacities of the combustion products and working mixture at constant volume in kJ/(kmole K).

Since the quantity of the residual gases M_r is insignificant as compared with the amount of the combustible mixture M_1, it may be assumed that the heat capacity $mc_v' = mc_v$, where mc_v is the mean kilomolar heat capacity of air in kJ/(kmole K).

The work of gas expansion in the section $z'z$ is

$$L_{z'z} = p_z V_z - p_{z'} V_{z'}$$

Since $p_{z'} = \lambda p_c$ and $V_{z'} = V_c$, it may be written that

$$L_{z'z} = p_z V_z - \lambda p_c V_c$$

Let us write the characteristic equations for the state of combustion products at the points z and c:

$$p_z V_z = 8314 (M_2 + M_r) T_z$$

and

$$p_c V_c = 8314 (M_1 + M_r) T_c$$

Inserting the values of work at the points z and c into the formula for determining the work $L_{z'z}$ (in kJ), we obtain

$$L_{z'z} = 8314 (M_2 + M_r) T_z - 8314\lambda (M_1 + M_r) T_c$$

After substituting the values of U_z, U_c and $L_{z'z}$ in equation (3.6) we have

$$\xi H_u = mc_v'' (M_2 + M_r) T_z - mc_v' (M_1 + M_r) T_c + \\ + 8314 (M_2 + M_r) T_z - 8314\lambda (M_1 + M_r) T_c$$

Dividing the equation termwise by the sum $M_1 + M_r$ and also cancelling the terms and transferring the known values to the left side of the equation, we get

$$\frac{\xi H_u}{M_1 + M_r} + (mc_v' + 8314\lambda) \frac{M_1 + M_r}{M_1 + M_r} T_c = (mc_v'' + 8314) \frac{M_2 + M_r}{M_1 + M_r} T_z$$

But

$$M_1 + M_r = M_1 \left(1 + \frac{M_r}{M_1}\right) = M_1 (1 + \gamma_r)$$

and the ratio $\frac{M_2 + M_r}{M_1 + M_r} = \mu$.

Inserting their values and also the value of kilomolar heat capacity $mc_p'' = mc_v'' + 8314$ into the equation found above, the final equation for the combustion in a compound cycle can be obtained:

$$\frac{\xi H_u}{M_1 (1 + \gamma_r)} + (mc_v' + 8314\lambda) T_c = \mu mc_p'' T_z \qquad (3.7)$$

From this equation the maximum temperature of the cycle of a compression-ignition engine (diesel) is estimated.

Since work of gases $L_{z'z} = 0$ for the cycle with combustion of liquid fuel at constant volume (carburettor engine), the equation of combustion may be simplified and, assuming the coefficient $\alpha \geqslant 1$, it takes the following form:

$$\frac{\xi H_u}{M_1 (1 + \gamma_r)} + mc'_v T_c = \mu mc''_v T_z \tag{3.8}$$

At some carburettor engine duties the coefficient $\alpha < 1$ and for this reason the fuel is not burned completely due to lack of oxygen. The heat loss in combustion of 1 kg of fuel due to lack of oxygen when $\alpha < 1$ is denoted by ΔH_u. Therefore, the difference $H_u - \Delta H_u$ must be substituted for H_u in the equation of combustion (3.8).

The value of ΔH_u (in kJ/kg) can be found from the formula:

$$\Delta H_u = 119{,}750 (1 - \alpha) L_0$$

Thus, the equation of combustion for a cycle with combustion at constant volume and $\alpha < 1$ may be written in the following form:

$$\frac{\xi (H_u - \Delta H_u)}{M_1 (1 + \gamma_r)} + mc'_v T_c = \mu mc''_v T_z \tag{3.9}$$

In equations (3.7) to (3.9) all the values are known except the temperature at the end of combustion, T_z, and the mean kilomolar heat capacities of the air and the combustion products with constant volume (mc'_v, mc''_v) and constant pressure (mc''_p), which can be found in reference books.

Since

$$mc'_v = a + bT$$
$$mc''_v = a''_v + b''_v T$$
$$mc''_p = a''_p + b''_p T$$

the equation of combustion for a compound cycle may be written in the following form:

$$\frac{\xi H_u}{M_1 (1 + \gamma_r)} + (a + bT_c) T_c + 8314 \lambda T_c = \mu (a''_p + b''_p T_z) T_z$$

Denoting the term $\mu b''_p$ by the letter A, the term $\mu a''_p$ by the letter B and the numerical value of all known magnitudes in the left side of the equation by the letter C, we can write the quadratic equation:

$$AT_z^2 + BT_z - C = 0$$

hence

$$T_z = \frac{-B + \sqrt{B^2 + 4AC}}{2A} \tag{3.10}$$

In this manner the combustion equation for a carburettor engine can be obtained:

$$\frac{\xi H_u}{M_1 (1 + \gamma_r)} + (a + bT_c) T_c = \mu (a''_v + b''_v T_z) T_z \tag{3.11}$$

Denoting the term $\mu b''_v$ by the letter A, the term $\mu a''_v$ by the letter B and the numerical value of all known magnitudes in the left side of the equation by the letter C, we obtain a similar quadratic equation for estimating the temperature T_z at the end of visible combustion.

The pressure at the end of combustion can be determined for any cycle from the two characteristic equations compiled for the end of combustion (point z) and the end of compression (point c):

$$p_z V_z = 8314 (M_2 + M_r) T_z \tag{3.12}$$

and

$$p_c V_c = 8314 (M_1 + M_r) T_c \tag{3.13}$$

Dividing the first equation by the second and after substitution

$$\frac{M_2 + M_r}{M_1 + M_r} = \mu, \quad \frac{p_z}{p_c} = \lambda \quad \text{and} \quad \frac{V_z}{V_c} = \rho$$

we get

$$\lambda = \frac{\mu}{\rho} \frac{T_z}{T_c}$$

Then, the pressure (in MN/m^2) at the end of combustion will be

$$p_z = \lambda p_c = \frac{\mu}{\rho} \frac{T_z}{T_c} p_c \tag{3.14}$$

for a compound cycle and

$$p_z = \mu \frac{T_z}{T_c} p_c \tag{3.15}$$

for a constant-volume cycle.

The coefficient ξ depends on the operating conditions of the engine, the method of mixture formation, the conditions of cooling the combustion chamber and afterburning on the expansion line.

The values of the coefficient ξ when an engine operates under a full load are illustrated below:

Carburettor engines	0.85-0.95
Gas engines	0.80-0.85
High-speed diesel engines .	0.65-0.85

The low values of the coefficient ξ indicate increased heat rejection to the cooling medium and considerable afterburning during expansion.

In the case of carburettor engines, the value of the coefficient ξ is selected with regard to the dissociation of the combustion products. Dissociation (splitting) of the CO_2 and H_2O molecules present in the combustion products occurs with absorption of heat and mainly depends on the temperature and pressure, increasing with a rise in temperature and diminishing with an increase in pressure.

Appreciable dissociation is observed when the excess air coefficient is approximately equal to 1, also with supercharging and at temperatures higher than 2300 K. During expansion the temperature of the gases falls and some of the heat expended on dissociation is released again due to oxidation of the elements formed during dissociation. However, this heat is of little use in the engine working process.

Dissociation is neglected in thermal calculations of diesel or gas engines because they operate with high excess air coefficients, rather low temperatures and high combustion pressures.

The increase in pressure λ lies within 3-4 for carburettor engines, 3-5 for gas engines and 1.2-1.4 for diesel engines.

Given below are the values of temperature T_z and pressure p_z of gases at the end of combustion when the engines operate under a full load:

	p_z, MN/m²	T_z, K
Carburettor engines . . .	3.0-5.0	2300-2700
Gas engines	2.5-4.5	2200-2500
Diesel engines	5.0-11.0	1800-2200

For diesel engines the pre-expansion ratio ρ lies within 1.2-1.7.

3.4. Process of Expansion

During the expansion process heat is converted into useful mechanical work.

As distinct from the theoretical cycle in which the expansion process occurs adiabatically, the actual process is attended by intensive heat exchange between the gases, on the one hand, and the walls of the cylinder, the combustion chamber and the piston crown, on the other, and also by a certain leakage of gas through loose joints. Besides, the gases receive heat as a result of afterburning, restoration of some dissociation products and the drop in the gas heat capacity due to the temperature reduction during expansion. As a result, the actual process of gas expansion occurs

by a polytropic curve with a variable exponent n_2 which changes within 1.15-1.30.

A rise in the temperatures of combustion and expansion causes the heat capacity of the gases to increase and the exponent n_2 to decrease. A higher engine cooling rate helps the exponent n_2 to increase.

It is assumed, when estimating the pressure and temperature at the end of the expansion process in a constant-volume cycle, that expansion occurs from TDC to BDC. For a compound cycle expansion begins from the end of heat delivery, with pressure being constant (point z), and ends at BDC.

According to the polytropic process equation $pV^n = \text{const}$,

$$p_b V_b^{n_2} = p_z V_z^{n_2}$$

We use this equation to find the final expansion pressure:

$$p_b = p_z \left(\frac{V_z}{V_b}\right)^{n_2}$$

Multiplying the numerator and denominator in the brackets by V_c, we obtain

$$p_b = p_z \left(\frac{V_z}{V_c}\frac{V_c}{V_b}\right)^{n_2}$$

But since

$$V_b = V_a, \quad \frac{V_z}{V_c} = \rho \quad \text{and} \quad \frac{V_c}{V_a} = \frac{1}{\varepsilon}$$

then

$$p_b = \left(\frac{\rho}{\varepsilon}\right)^{n_2} p_z$$

Remembering that

$$\frac{V_b}{V_z} = \frac{V_a}{V_z} = \frac{\frac{V_a}{V_c}}{\frac{V_z}{V_c}} = \frac{\varepsilon}{\rho} = \delta$$

i.e., it represents the *subsequent expansion ratio*, we shall obtain for a compound cycle

$$p_b = \frac{p_z}{\delta^{n_2}} \tag{3.16}$$

For a constant-volume cycle

$$V_z = V_c, \quad \frac{V_b}{V_z} = \frac{V_a}{V_c} = \varepsilon = \delta$$

since $\rho = 1$.

Substituting the found value of δ into formula (3.16)

$$p_b = \frac{p_z}{\varepsilon^{n_2}} \tag{3.17}$$

At the beginning of expansion, the inflow of heat to the gases due to intensive afterburning is much larger than the heat rejection to the cooling medium (to the walls). For this reason, the polytropic combustion exponent n_2 (Fig. 3.9) is less than the adiabatic expansion exponent k (section zm). As afterburning reduces, the value of the polytropic exponent n_2 increases. If the heat evolved during afterburning is equal to the heat losses to the walls, the polytropic curve will intersect the adiabatic line at the point H.

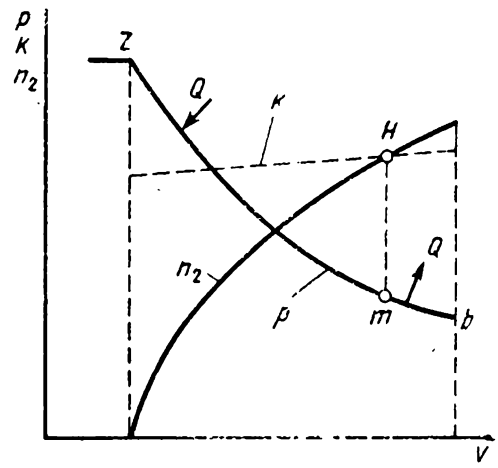

Fig. 3.9. Mutual arrangement of expansion curves (z-b) and adiabatic k and polytropic n_2 expansion exponents

As the gases continue to expand, the heat losses to the walls are larger than the inflow of heat from afterburning and the value of the exponent n_2 increases (section mb).

Since it is extremely difficult to take into account all the phenomena accompanying the process of expansion, the actual curve of expansion with a variable exponent is replaced, as in the process of compression, by a polytropic curve with a constant exponent n_2.

The mean value of the polytropic expansion exponent n_2 is less than the adiabatic exponent k and for this reason the curve of the polytropic expansion exponent is arranged above the curve of the adiabatic exponent.

The mean values of the polytropic expansion exponent n_2 are illustrated below:

Carburettor engines	1.25-1.33
Gas engines	1.25-1.35
Diesel engines	1.22-1.25

A number of factors affect the mean value of the exponent n_2. A higher engine crankshaft angular velocity tends to decrease the exponent n_2, this can be attributed to considerable afterburning and reduction of heat rejection to the walls.

The temperature at the end of expansion can be found from the equality obtained by a joint solution of the characteristic and polytropic equations:

$$T_b V_b^{n_2-1} = T_z V_z^{n_2-1}$$

Having solved this equation with respect to the temperature T_b we obtain

$$T_b = T_z \left(\frac{V_z}{V_b}\right)^{n_2-1}$$

For a compound cycle

$$T_b = \frac{T_z}{\delta^{n_2-1}} \tag{3.18}$$

and for a constant-volume cycle

$$T_b = \frac{T_z}{\varepsilon^{n_2-1}} \tag{3.19}$$

Given below are the values of pressure p_b and temperature T_b at the end of the expansion process:

	p_b, MN/m^2	T_b, K
Spark-ignition engines . .	0.4-0.6	1300-1700
Diesel engines	0.3-0.5	1000-1200

The reduction in the pressure and temperature values at the end of expansion in diesel engines can be explained by the higher gas expansion ratio and lower temperature at the end of combustion.

3.5. Process of Exhaust

The exhaust process should reduce to the minimum the pressure p_r of the residual gases at the end of the exhaust stroke and the expenditure of energy on this process.

In modern engines the exhaust valve opens early, this reduces the work energy expended in exhaust and improves used gas scavenging in the cylinder. However, this tends to reduce slightly the engine power because the duration of the expansion process is shortened.

The pressure of the used gases in the cylinder never remains constant during exhaust because of elastic oscillations of gases occurring in the exhaust system due to the periodic nature of this process. Since the oscillations of gases during exhaust cannot be calculated theoretically, the pressure in the process of exhaust is usually taken constant and equal to a certain mean value during the entire process.

The exhaust pressure p_r depends on the angular velocity of the engine crankshaft, the design and dimensions of the exhaust system and lies within 0.105-0.12 MN/m^2.

The temperature of the exhaust gases mainly depends on the same factors as the temperature at the end of the expansion process. Maximum temperature of the used gases is obtained with a

slightly weak mixture, i.e., approximately with the coefficient $\alpha = 1.05\text{-}1.15$. Further weakening of the mixture reduces the temperature of the used gases, because the maximum temperature of the cycle falls despite the greater duration of combustion.

Temperatures at the end of exhaust T_r (in K) are illustrated below:

Carburettor engines	900-1100
Diesel engines	700-900

The waste gases possess considerable energy which, when used, improves the fuel economy of a reciprocating engine. The energy of the waste gases can be utilized to drive the gas turbine of a reciprocating engine centrifugal supercharger.

The waste gases utilized in the gas turbine increase the resistance in the exhaust system and the pressure p_r. However, this resistance growth is completely compensated for by the higher intake pressure.

The composition of the waste gases depends on the quality of the working mixture, the methods of forming and igniting the mixture, operating conditions, the technical state of the engine, etc. With a well developed design and an engine in good repair the composition of the waste gases mainly depends on the amount of oxygen present in the engine combustion chambers.

Apart from carbon dioxide, water vapour and oxygen, the exhaust gases contain toxic substances. The principal components which determine the toxic action of the exhaust gases are carbonic and nitric oxides and aldehydes.

It has been established that the content of carbonic oxide in the exhaust gases of carburettor engines is maximum at idle run and also with small and maximum loads on the engine. This corresponds to rich mixture operation ($\alpha < 1$), i.e., with considerably incomplete combustion of fuel.

The carbonic oxide content in the exhaust gases of carburettor engines is often increased by a wrong carburettor and ignition adjustment, by using low octane petrol and engine wear (worn-out piston rings, pistons and walls of cylinders).

The exhaust gases of carburettor engines contain 0.02-1 per cent of carbonic oxide by volume. It should be borne in mind that as the car moves the exhaust gases are mixed rapidly with the surrounding air and prevent local gas-filled areas. The exhaust gases of high-speed diesel engines contain 2-3 times less carbonic oxide than carburettor engines. This can be attributed to the higher excess air coefficient and a lower fuel combustion temperature.

Nitric oxides in the exhaust gases comprise 0.001-0.2 per cent by volume. The formation of nitric oxides is stimulated by the

temperature increase during combustion and a rise in the compression ratio of carburettor engines.

The principal methods of reducing the toxic action of the exhaust gases are as follows.

Engines, their separate units and systems (especially the systems of delivery and ignition) should operate in a stable regime.

It has been found that when light fuels are injected into the cylinders of a diesel engine, less carbonic and nitric oxides are formed in the exhaust gases than in carburettor engines. This can be explained by the comparatively slow combustion process and decreased process temperatures. The diesel engines operating on heavy and light fractions of petroleum fuels (petrols, etc.) are known as *multi-fuel* ones. Therefore, the use of light fuels in diesel engines reduces the toxic action of the exhaust gases. The toxic action of the exhaust gases in engines with precombustion-chamber ignition is much less (see below) because the mixture is burned in these engines with higher excess air coefficient values. The use of liquefied gases (propane-butane mixtures) for engines also helps to reduce appreciably the toxic action of exhaust gases.

Afterburners are an effective means in reducing the toxic action of the exhaust gases. There are catalytic and flame afterburners. A catalytic afterburner is installed inside a housing near the engine exhaust pipe. A thin layer of catalyst made of a platinum-aluminium alloy, vanadium or cobalt oxide is placed on ceramic bars, balls or lattices. The exhaust gases pass together with sucked-in air through the catalyst which ensures afterburning of the combustible toxic substances present.

Flame after burners are employed to lower the toxic action of diesel engine exhaust gases. In this case, the exhaust gases are heated in a separate chamber by a special burner operating on liquid fuel. The higher temperature of the exhaust gases promotes burning of the toxic admixtures.

Chapter IV BASIC PARAMETERS AND ENGINE EFFICIENCY

4.1. Mean Indicated Pressure, Indicated Power and Fuel Economy

In order to determine the indicated power of an engine, the *mean indicated pressure* p_i must be known. The pressure p_i is the conditional constant pressure of gases, under the action of which work equal to the indicated work of the gases during the entire cycle is performed in expansion (Fig. 4.1).

If the indicated work of a cycle in MN·m is denoted by L_i' and the cylinder working volume in m^3 by V_h, the mean indicated pressure (in MN/m^2) will be

$$p_i' = \frac{L_i'}{V_h} \qquad (4.1)$$

It follows from equation (4.1) that the mean indicated pressure is the specific indicated work obtained from a unit of the working volume of the cylinder.

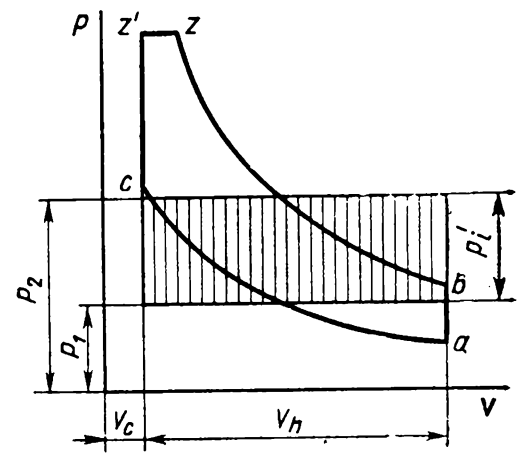

Fig. 4.1. Theoretical indicator diagram and mean indicated pressure

In graphical form, the pressure p_i' represents the height of a rectangle with base V_h and an area equal to the area of the indicator diagram. The pressure p_1 is the mean geometrical pressure during the process of compression and the pressure p_2 is the mean geometrical pressure during the combustion and expansion processes.

Let us first determine the mean indicated pressure of a compound cycle, and then of a constant-volume cycle.

The work of gases L_i' (in MN·m) per cycle is equal to the difference between useful work $L_{z'z} + L_{zb}$ expended on the combustion and expansion processes and the work L_{ac} used for compression:

$$L_i' + L_{z'z} + L_{zb} - L_{ac} \qquad (4.2)$$

The work in the section $z'z$ of pre-expansion at constant pressure can be found, according to Fig. 4.1, from the expression:

$$L_{z'z} = p_z (V_z - V_{z'}) = p_z V_{z'} \left(\frac{V_z}{V_{z'}} - 1 \right)$$

Since

$$V_{z'} = V_c, \quad p_z = \lambda p_c \quad \text{and} \quad \frac{V_z}{V_c} \rho$$

then

$$L_{z'z} = \lambda (\rho - 1) p_c V_c$$

The work of polytropic expansion of gases in the section zb is

$$L_{zb} = \frac{1}{n_2 - 1} (p_z V_z - p_b V_b) = \frac{p_z V_z}{n_2 - 1} \left(1 - \frac{p_b V_b}{p_z V_z}\right)$$

Since

$$\frac{p_b V_b}{p_z V_z} = \frac{T_b}{T_z} = \left(\frac{V_z}{V_b}\right)^{n_2 - 1} = \frac{1}{\delta^{n_2 - 1}}$$

and

$$p_z V_z = \lambda \rho p_c V_c$$

then

$$L_{zb} = \frac{\lambda \rho}{n_2 - 1} \left(1 - \frac{1}{\delta^{n_2 - 1}}\right) p_c V_c$$

The work of polytropic compression in the section ac

$$L_{ac} = \frac{1}{n_1 - 1} (p_c V_c - p_a V_a) = \frac{p_c V_c}{n_1 - 1} \left[1 - \left(\frac{V_c}{V_a}\right)^{n_1 - 1}\right] =$$

$$= \frac{1}{n_1 - 1} \left(1 - \frac{1}{\varepsilon^{n_1 - 1}}\right) p_c V_c$$

Inserting the found expressions, which determine the work $L_{z'z}$, L_{zb} and L_{ac}, into equation (4.2) and taking the product $p_c V_c$ outside the brackets, we obtain

$$L_i' = p_c V_c \left[\lambda (\rho - 1) + \frac{\lambda \rho}{n_2 - 1} \left(1 - \frac{1}{\delta^{n_2 - 1}}\right) - \frac{1}{n_1 - 1} \left(1 - \frac{1}{\varepsilon^{n_1 - 1}}\right)\right]$$

Substituting this expression in formula (4.1) and replacing the ratio of the volumes $\frac{V_c}{V_h}$ by $\frac{1}{\varepsilon - 1}$, we get the final formula which estimates the mean indicated pressure (in MN/m²) of the compound cycle:

$$p_i' = \frac{p_c}{\varepsilon - 1} \left[\lambda (\rho - 1) + \frac{\lambda \rho}{n_2 - 1} \left(1 - \frac{1}{\delta^{n_2 - 1}}\right) - \frac{1}{n_1 - 1} \left(1 - \frac{1}{\varepsilon^{n_1 - 1}}\right)\right] \tag{4.3}$$

For a constant-volume cycle ($\rho = 1$ and $\delta = \varepsilon$)

$$p_i' = \frac{p_c}{\varepsilon - 1} \left[\frac{\lambda}{n_2 - 1} \left(1 - \frac{1}{\varepsilon^{n_2 - 1}}\right) - \frac{1}{n_1 - 1} \left(1 - \frac{1}{\varepsilon^{n_1 - 1}}\right)\right] \tag{4.4}$$

The mean indicated pressures of the actual cycles of four-stroke engines, calculated by formulas (4.3) and (4.4), have smaller values due to the rounded-off portions of the indicator diagram and pumping losses Δp_i arising during admission and exhaust.

The indicator diagram rounding-off losses are taken into account by the *diagram rounding-off coefficient* φ_r which lies within 0.92-0.97 for four-stroke engines. The values of the coefficient φ_r for carburettor and gas engines are larger than for diesel engines.

Therefore, the actual mean indicated pressure (in MN/m²) of four-stroke engines is

$$p_i = \varphi_r p_i' - \Delta p_i \tag{4.5}$$

where Δp_i is the pumping losses.

The pumping losses in four-stroke engines can approximately be determined from the formula:

$$\Delta p_i = p_r - p_a$$

where p_r = pressure at the end of exhaust, MN/m²
p_a = pressure at the end of admission, MN/m²

According to the experimental data, the value of Δp_i for high-speed engines without supercharging lies within 0.025-0.040 MN/m².

The pumping losses belong to mechanical losses, since in practice the mechanical losses, i.e., losses due to friction, are usually estimated on a test stand by motoring a hot engine by the stand electric motor. With this method of determining mechanical losses, the power spent on motoring also includes the power required for the admission and exhaust of air, i.e., the power of the pumping losses. For this reason, formula (4.5) takes the form

$$p_i = \varphi_r p_i' \tag{4.6}$$

If the line of admission is arranged above the line of exhaust ($p_a > p_r$), for example, when using a supercharger, the work expended on auxiliary processes is also accounted for by the value of mechanical losses in the engine.

In the case of two-stroke engines, the actual mean indicated pressure is determined with regard to the *part of lost volume* ψ and the diagram rounding-off coefficient φ_r:

$$p_i = \varphi_r (1 - \psi) p_i' \tag{4.7}$$

The coefficient φ_r for two-stroke diesel engines depends on the type of scavenging and lies within 0.92-1.0. The lower limit refers to the case of uniflow scavenging and exhaust of gases through the valves, and the upper one for exhaust and scavenging effected through the ports.

The values of mean indicated pressure p_i (in MN/m²) of four-stroke engines under a full load are specified below:

Carburettor engines	0.7-1.2
Gas engines	0.6-0.9
Diesel engines:	
without supercharging	0.6-1.0
supercharged	0.8-2.0 and over

If the indicator diagram of the actual cycle of an engine is made, the mean indicated pressure can be found from the formula:

$$p_i = \frac{F}{l} m_a$$

where F = area of the indicator diagram, mm²
l = length of the indicator diagram, mm
m_a = scale of pressures in the indicator diagram, MN/(m²·mm)

The indicated power is the work performed by gases in the cylinder of an engine in unit time.

The indicated work in MN·m done by gases in the cylinder for one cycle

$$L_i = p_i V_h$$

The number of working cycles occurring in the cylinder per second is equal to $\frac{2n}{60\tau} = \frac{\omega_e}{\pi\tau}$ (where n is the crankshaft speed in rev/min and τ is the number of piston strokes per cycle). Therefore, the power developed in one cylinder is

$$N_i' = \frac{p_i V_h \omega_e}{\pi\tau}$$

Then, the indicated power of a multi-cylinder engine with i_{cyl} cylinders can be found from the formula:

$$N_i = \frac{p_i V_h \omega_e i_{cyl}}{\pi\tau} \qquad (4.8)$$

where the pressure p_i is in kN/m², volume V_h in litres and angular velocity ω_e in rad/s.

The indicated efficiency is the ratio of the heat converted into useful work L_i of a cycle to all expended heat H_u:

$$\eta_i = \frac{L_i}{H_u}$$

Therefore, the indicated efficiency is used to estimate the degree of heat utilization in the cylinder of an engine with account of all thermal losses including heat rejection to the cool sink.

As shown above, the work of a cycle is equal to the product $p_i V_h$; the volume V_h can be determined from the characteristic equation:

$$\eta_V p_0 V_h = 8314 M_1 T_0$$

solving it with respect to the volume V_h

$$V_h = \frac{8314 M_1 T_0}{\eta_V p_0}$$

where the pressures p_i and p_0 are in N/m^2, volume V_h in m^3 and $M_1 = \alpha L_0$.

Inserting the expressions for V_h into the formula for determining the work L_i we find that

$$L_i = \frac{8314 p_i M_1 T_0}{\eta_V p_0}$$

Inserting the expression L_i into the formula for estimating the coefficient η_i we obtain

$$\eta_i = 8314 \frac{p_i M_1 T_0}{\eta_V H_u p_0} \tag{4.9}$$

If the indicated power N_i in kW, hourly consumption of fuel G_m in kg/hr and its heat of combustion H_u in kJ/kg are known, the indicated efficiency can be determined from the formula:

$$\eta_i = \frac{3600 N_i}{G_m H_u}$$

or

$$\eta_i = \frac{3600}{g_i H_u} \tag{4.10}$$

where 3600 kJ/(kW·hr) = thermal equivalent of work equal to 1 kW·hr

$g_i = \frac{G_f}{N_i}$ = specific indicated fuel consumption in kg/(kW·hr)

Given below are the values of the indicated efficiency η_i:

Carburettor engines	0.25-0.35
Gas engines	0.28-0.35
Diesel engines	0.38-0.50

The values of the indicated fuel consumption g_i in g/(kW·hr) are:

Carburettor engines	250-340
Diesel engines	180-230

4.2. Effective (Brake) Power and Engine Fuel Economy

The indicated power developed in the cylinder of an engine cannot be utilized completely. Part of this power is expended on mechanical losses in overcoming friction between the rubbing members of the engine (between the pistons and cylinders, the crankshaft and its bearings, etc.) and also between the moving members (flywheel, connecting rods, cranks of the shaft, etc.) and the air. Some power goes for pumping losses, and to drive auxiliary mechanisms (pumps, fans, generator, supercharger, etc.).

For this reason, the *effective power of an engine* N_e (in kW) taken off the flywheel is less than the indicated power N_i by the amount of *power needed to overcome the mechanical losses* N_m and can be found from the expression:

$$N_e = N_i - N_m \tag{4.11}$$

The power accounting for the mechanical losses (in kW) can be estimated as well as the indicated power, i.e., from the formula:

$$N_m = \frac{p_m V_h \omega_e i_{cyl}}{\pi\tau}$$

where p_m is the mean pressure of the mechanical losses, i.e., part of the mean indicated pressure expended on mechanical losses, kN/m^2.

This formula can be employed to find the *mean pressure of the mechanical losses*:

$$p_m = \frac{\pi\tau N_m}{V_h \omega_e i_{cyl}} 10^{-3} \tag{4.12}$$

It has been established experimentally that the pressure p_m depends on the type of the engine, piston speed, number and diameter of the cylinders, ratio of the piston stroke to the cylinder diameter S/D and load on the engine. The pressure p_m changes depending on the angular velocity ω_e by the linear law both in carburettor and diesel engines:

$$p_m = a + bc_p \tag{4.13}$$

where $c_p = \frac{S\omega_e}{\pi}$ = mean piston speed, m/s

S = piston stroke, m

a and b = coefficients which depend on the factors indicated above

The values of the coefficients a and b are specified below:

	a	b
Carburettor engines	0.035	0.012
Diesel engines:		
with open combustion chambers	0.113	0.007
with divided combustion chambers	0.113	0.010

If the mean pressure of mechanical losses p_m is subtracted from the mean indicated pressure p_i the remaining mean pressure represents the conditional constant pressure on the piston, in which case the work is created equal to the effective work on the engine shaft. This value is known as the *mean effective pressure*:

$$p_e = p_i - p_m \qquad (4.14)$$

The values of pressures p_i, p_m and p_e are illustrated in Fig. 4.2 which also shows the work of the mechanical losses L_m.

The brake power N_e of the engine can be expressed, by analogy with the indicated power N_i, through the mean effective pressure p_e:

$$N_e = \frac{p_e V_h \omega_e i_{cyl}}{\pi\tau} \qquad (4.15)$$

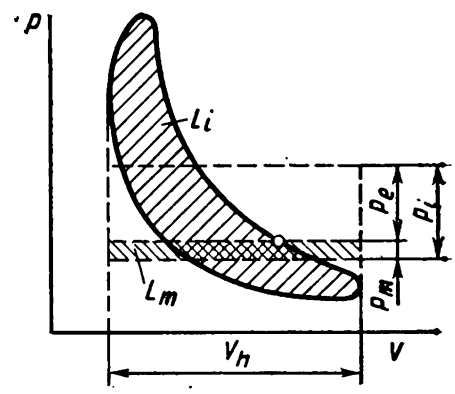

Fig. 4.2. Graphical representation of the values of p_i, p_m and p_e

Solving equation (4.15) with respect to the mean effective pressure (in MN/m²), we obtain

$$p_e = \frac{\pi\tau N_e}{V_h \omega_e i_{cyl}} 10^{-3} \qquad (4.16)$$

The mechanical efficiency of an engine which accounts for the mechanical losses is the ratio of the brake power to the indicated one:

$$\eta_m = \frac{N_e}{N_i} = \frac{p_e}{p_i}$$

Since $N_e = N_i - N_m$ and $p_e = p_i - p_m$ the formula for the mechanical efficiency may be written in another form:

$$\eta_m = 1 - \frac{N_m}{N_i} = 1 - \frac{p_m}{p_i} \qquad (4.17)$$

It follows from formulas (4.13), (4.15) and (4.17) that in raising the crankshaft angular velocity the brake power and the mean pressure of the mechanical losses increase, and the mechanical

efficiency (with a constant mean indicated pressure p_i) decreases. This occurs due to the growth in the forces of inertia and the increase in the mechanical losses.

When the load and, therefore, the pressure p_i decreases the values of the mechanical efficiency are reduced, and at idle run, when the pressure $p_i = p_m$, the coefficient $\eta_m = 0$. In the latter case all the indicated power is expended to compensate for the mechanical losses, i.e., $N_i = N_m$, and for this reason the brake power of the engine is also equal to zero.

Given below are the values of the mechanical efficiency η_m at the maximum power of four-stroke engines:

Carburettor engines	0.70-0.85
Gas engines	0.75-0.85
Diesel engines	0.7-0.8

The values of the mean effective pressure p_e (in MN/m^2) at the maximum power of four-stroke engines are as follows:

Carburettor engines	0.5-1.0
Gas engines	0.45-0.75
Diesel engines	0.5-0.8

The engine operation economy as a whole is estimated by the specific fuel consumption g_e per unit of the effective work (1 kW·hr).

The brake thermal efficiency is the ratio of heat converted into useful work on the engine shaft to all expended heat:

$$\eta_e = \frac{3600 N_e}{G_f H_u} \tag{4.18}$$

or

$$\eta_e = \frac{3600}{g_e H_u} \tag{4.19}$$

where $g_e = \frac{G_f}{N_e}$ is the specific fuel consumption in kg/(kW·hr).

Hence, the brake thermal efficiency η_e takes into account both heat and mechanical losses in the engine and is, therefore, equal to the product $\eta_i \eta_m$.

The values of the brake thermal efficiency η_e at full power are illustrated below:

Carburettor engines	0.24-0.30
Gas engines	0.22-0.28
Diesel engines	0.30-0.38

It may be written on the basis of the above formulas that

$$\frac{g_i}{g_e} = \frac{\eta_e}{\eta_i} = \eta_m$$

whence

$$g_e = \frac{g_i}{\eta_m} \tag{4.20}$$

Given below are the values of the specific fuel consumption g_e in g/(kW·hr):

Carburettor engines	290-380
Diesel engines	230-280

The specific fuel consumption g_e is inversely proportional to the brake thermal efficiency η_e and depends on the speed and load operating conditions of the engine. At stable duty (ω_e = const) the minimum fuel consumption value g_e corresponds to the maximum value of the coefficient η_e. This is called *economical* running.

When the load is decreased, the specific fuel consumption in carburettor engines sharply increases, since in this case the values of the coefficients η_i and η_m diminish. If the engine idles the coefficient η_m is equal to zero and the effective fuel consumption g_e, to infinity.

If the load is increased to 80-85 per cent of the full load the fuel economy of carburettor engines is worsened. This is caused by a drop in the coefficient η_m and the engagement of the carburettor economizer.

As the load drops in diesel engines, the coefficient α increases because less fuel is injected. This increases the coefficient η_i and decreases the specific indicated fuel consumption g_i.

The specific fuel consumption g_i in diesel engines slightly increases when the load is decreased due to the reduction in the coefficient η_m. As the load on a diesel engine is increased this consumption grows because the fuel combustion process is worsened. The load on a diesel engine may be increased only until soot deposits (limit of smoking) appear in the exhaust gases.

Power-per-Litre of an Engine

The operation of an engine from the standpoint of utilizing the working volume and also its thermal and dynamic stress can be evaluated by *power-per-litre.*

The power-per-litre of an engine N_{lit} is the maximum brake power as referred to one litre of the engine working volume:

$$N_{lit} = \frac{N_{e\,max}}{V_h i_{cyl}} \tag{4.21}$$

The power-per-litre value makes it possible to compare and estimate the perfection of the working processes and designs of engines of various types and models. The higher the power-per-

litre, the smaller, all other conditions being equal, the overall dimensions and mass of the engine.

The values of the power-per-litre N_{lit} (in kW/lit) are illustrated below:

Carburettor engines:	
mass-produced cars . .	15-37
sport-racing cars	60-110
Diesel engines	11-22

The working volume of the cylinder (in lit) is

$$V_h = \frac{\pi D^3}{4} \cdot \frac{S}{D} \tag{4.22}$$

The $\frac{S}{D}$ ratio for up-to-date motor vehicle engines varies from 0.85 to 1.0. It comes to 1.4 in engines of an old design.

The $\frac{S}{D}$ ratio has an appreciable effect on the overall dimensions and mass of the engine, as well as on the course of the working process. An increase in the $\frac{S}{D}$ ratio is attended by the growth in the mean speed of the piston, which increases the forces of inertia and decreases, therefore, the value of the mechanical efficiency and intensifies wear of the engine rubbing parts. Besides, a greater ratio increases the height and mass of the engine and deteriorates charging of the cylinders and the indicated efficiency. However, a higher ratio reduces the cylinder diameter and the probability of detonation.

A reduction in the $\frac{S}{D}$ ratio decreases the mean speed of the piston, the height and mass of the engine, improves the admission and increases the indicated efficiency, but the lower ratio tends to increase the diameter of the piston and also the force of gases exerted on the piston.

4.3. Heat Balance of an Engine

It was shown when investigating the cycle of an internal-combustion engine that only part of the heat liberated during fuel combustion in the cylinder is converted into useful work. The rest of the heat is dissipated in thermal losses.

The distribution of the heat quantity evolved during combustion into useful work and various losses is characterized by the *engine external heat balance.* The heat balance is compiled from the results of tests with a settled heat state of the engine for various operating conditions.

The value of each heat balance component is referred to unit time (1 hr) and is determined in per cent to the entire quantity of the available heat.

The heat balance may be used to establish the degree of engine perfection and point out ways of eliminating the causes which impair its economical work.

The heat balance equation may be written in the following form:

$$Q = Q_e + Q_c + Q_g + Q_{ic} + Q_r \tag{4.23}$$

where Q = available heat liberated upon full combustion of fuel, kJ

Q_e = heat equivalent to the useful (effective) work of the engine

Q_c = heat rejected to the cooling medium

Q_g = heat carried away with the exhaust gases

Q_{ic} = heat which is not liberated due to incomplete combustion

Q_r = remainder of heat balance which includes all kinds of unaccounted-for losses

If the heat balance is compiled in per cent of the entire quantity of the available heat, its equation may be written in the following form:

$$q = q_e + q_c + q_g + q_{ic} + q_r = 100\%$$

where

$$q_e = \frac{Q_e}{Q} 100\%, \quad q_c = \frac{Q_c}{Q} 100\%, \text{ etc.}$$

The available amount of heat Q as referred to one hour can be found from the formula:

$$Q = G_f H_u$$

where G_f = hourly consumption of fuel, kg/hr

H_u = heat of fuel combustion, kJ/kg

The quantity of heat (in kJ/hr) equivalent to the useful work of the engine is

$$Q_e = 3600 N_e \tag{4.24}$$

or

$$Q_e = \eta_e Q$$

The amount of heat rejected to the cooling medium through the cylinder walls and head and also through the piston and other

parts with liquid cooling can be found from the formula:

$$Q_w = G_w (T_2 - T_1) \tag{4.25}$$

where G_w = hourly consumption of water passing through the cooling system, kg/hr

T_1 and T_2 = temperature of water at the inlet and outlet of the system, K

The cooling water does not only receive the heat from the gases but also the heat which is equivalent to the work of friction of the pistons in the cylinders. The amount of heat withdrawn by the oil for engines without oil cooling of the pistons reaches 2-3 per cent and with oil cooling — 8-12 per cent of the available heat.

The amount of heat (in kJ/hr) carried away with the exhaust gases is determined from the formula:

$$Q_g = G_f (M_2 m c''_p T_g - M_1 m c_p T_0) \tag{4.26}$$

where T_g and T_0 are the temperatures of the waste gases behind the exhaust pipe and of the fresh charge, K.

The quantity of heat not liberated due to incomplete combustion is estimated for the heat balance only when the coefficient $\alpha < 1$. If the coefficient $\alpha \geqslant 1$ this loss is small and is included in the remainder of the heat balance.

The loss of heat due to incomplete combustion can be found from the formula:

$$Q_{ic} = \Delta H_u G_f \tag{4.27}$$

where, as indicated above,

$$\Delta H_u = 119{,}750 (1 - \alpha) L_0 \text{ kJ/kg}$$

The remainder of the heat balance of an engine which is the total amount of heat of unaccounted-for losses can be determined as the difference

$$Q_r = Q - (Q_e + Q_c + Q_g + Q_{ic}) \tag{4.28}$$

The remainder of the balance includes the heat losses due to the heat radiation by the external surfaces of the engine, the heat corresponding to the work of friction except for the heat liberated in friction of the pistons against the walls of the cylinders, the heat corresponding to the kinetic energy of the exhaust gases, and the heat which has not been evolved due to incomplete vaporization of the fuel and its poor mixing with the air.

The values of the separate components of the engine heat balance are never constant but vary depending on the load, compression ratio, composition of the mixture, thermal state and angular

velocity of the engine crankshaft. Table 4.1 illustrates the approximate values of some components of the external heat balance of automobile engines at maximum power.

TABLE 4.1

Values of the Components of the External Heat Balance of Automobile Engines in %

Components of heat balance	Carburetor engines	Gas engines	Diesel engines
Heat converted into useful work	22-30	24-30	30-38
Heat rejected into cooling medium	12-25	20-25	15-35
Heat carried away by exhaust gases	30-35	35-45	25-45
Heat not evolved due to incomplete combustion	0-3	0-5	0-5
Heat of unaccounted-for losses	3-10	4-10	2-5

Figure 4.3 shows the graphs of the heat balance change depending on the angular velocity of the engine crankshaft with throttle opened wide (Fig. 4.3*a*) and on the composition of the mixture

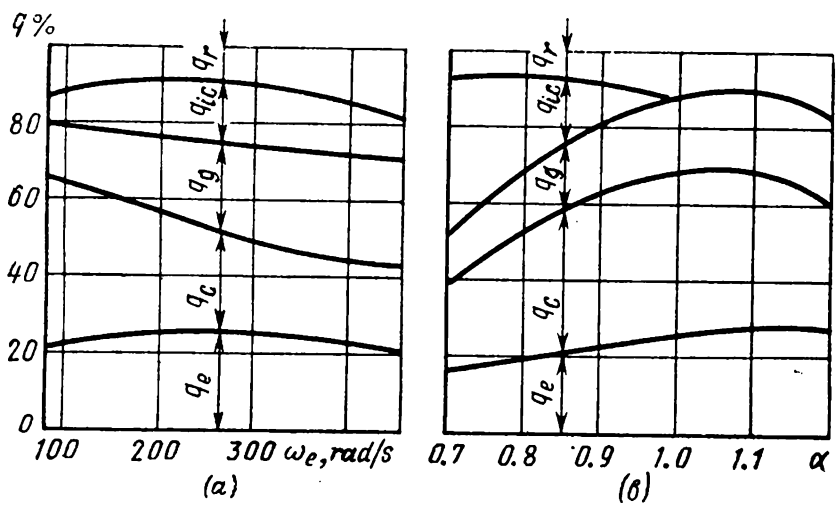

Fig. 4.3. Heat balance of an engine versus
(*a*) angular velocity of crankshaft ω_e; (*b*) composition of mixture (excess air coefficient α)

(Fig. 4.3*b*). It can be gathered from the graphs that a growth in the angular velocity ω_e increases the values of q_g and decreases the values of q_c because the time during which the heat of the exhaust gases is partly rejected to the cooling medium is reduced, and their temperature slightly increases. Besides, the best fuel economy of the engine (Fig. 4.3*b*) is obtained with the coefficient

$\alpha = 1.05\text{-}1.15$. If the coefficient $\alpha < 1$, the heat losses q_c and q_2 diminish but the heat losses q_{ic} increase due to incomplete combustion, which impairs the fuel economy of the engine.

4.4. Increasing the Power and Improving the Fuel Economy of the Engine

The designs of automobile engines have been developed today along the lines of increasing the power and improving the fuel economy, reducing the overall dimensions and weight and prolonging the service life.

The effective power (in kW) of an engine can be found from formula (4.15):

$$N_e = \frac{p_e V_h \omega_e i_{cyl}}{\pi\tau}$$

where

$$p_e = 10^{-3} \eta_m p_i$$

Inserting instead of p_i the value found from formula (4.9) and remembering that

$$M_1 = \alpha L_0 \quad \text{and} \quad \frac{p_0}{T_0} = R\rho_0$$

we get

$$p_e = 10^{-3} \frac{H_u}{L_0} \frac{\eta_i}{\alpha} \eta_V \eta_m \rho_0$$

where ρ_0 is the density of air in kg/m^3.

It follows from the above formulas for determining the power and the pressure p_e that the power N_e can be increased by changing the parameters in these formulas: for example, by increasing the working volume V_h and the number of cylinders i_{cyl}. However, this tends to increase the overall dimensions and mass of the engine. This method can be employed only if all the other methods of improving the engine power-per-litre have proved of no avail.

If the fuel economy of an engine is good with higher wear resistance the perfection of the design can be evaluated by its power-per-litre:

$$N_{lit} = \frac{p_e \omega_e}{\pi\tau} = \frac{10^{-3}}{\pi} \frac{H_u \eta_i 1}{L_0 \alpha \tau} \eta_V \rho_0 \omega_e \eta_m \qquad (4.29)$$

From formula (4.29) it is possible to analyze the effect of various factors on the value of the power N_{lit} and indicate ways for its improvement.

When selecting a method of increasing the power-per-litre, it is necessary to take into account the effect of all factors in the right side of equation (4.29).

The ratio $\frac{H_u}{L_0}$ for liquid fuels employed in up-to-date engines is within rather narrow limits; for this reason, the power N_{lit} practically remains constant when the ratio $\frac{H_u}{L_0}$ is changed. Therefore, the power-per-litre of an engine can be improved by increasing the values of $\frac{\eta_i}{\alpha}$, $\frac{1}{\tau}$, η_V, ρ_0, ω_e and η_m. All these values influence also the engine performance.

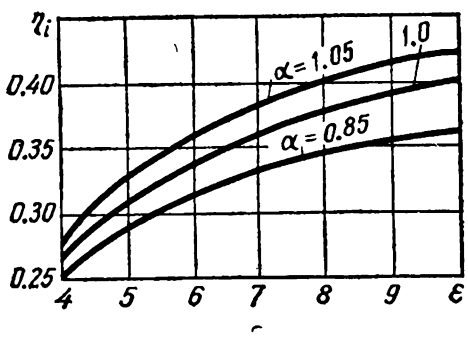

Fig. 4.4. Effect of compression ratio ε and excess air coefficient α on indicated efficiency

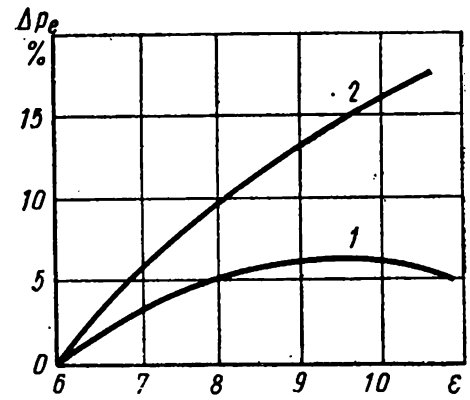

Fig. 4.5. Growth in pressure Δp_e versus compression ratio ε in engines with variously positioned valves
1—with side valve, *2*—with overhead valve

The ratio $\frac{\eta_i}{\alpha}$ describes the course of the engine actual cycle. The indicated efficiency mainly depends on the compression ratio and composition of the combustible mixture. When the compression ratio and the excess air coefficient are increased to definite values the coefficient η_i increases too (Fig. 4.4).

The coefficient $\alpha = 1.05\text{-}1.08$ corresponds to the maximum value of the coefficient η_i. The ratio $\frac{\eta_i}{\alpha}$ in carburettor engines attains its maximum when the coefficient $\alpha = 0.85\text{-}0.90$. In diesel engines the ratio $\frac{\eta_i}{\alpha}$ has its maximum when $\alpha > 1$, since the mixture formation process in these engines is less perfect than in carburettor engines: when $\alpha \leqslant 1$, the diesel cycle occurs with incomplete fuel combustion and is accompanied by the evolution of soot deposits in the exhaust gases.

The power-per-litre of carburettor engines is higher when the compression ratio ε is increased and the coefficient α is reduced. Better fuel economy can be obtained if the compression ratio ε

and the coefficient α are increased. Therefore, the compression ratio should be increased and the combustible mixture made weaker to promote power and improve fuel economy. This allows high compression ratios to be used with the maximum advantage.

The work of a carburettor engine on lean mixtures can be ensured by installing two carburettors operating consecutively. During operation at small and medium loads the mixture is taken from the carburettor adjusted for feeding the lean mixture. In the case of heavy loads the work is done by the second carburettor adjusted to give a rich mixture. However, if the mixture is ignited by a spark plug with the usual (non-electronic) ignition system, it is hardly possible to ensure continuous ignition of a very lean mixture. A carburettor engine can operate with lean working mixture if fuel spray ignition is used (see Chapter VI).

Engine power can be increased if the two-stroke cycle ($\tau = 2$) is employed. Theoretically, all other conditions being equal, the two-stroke cycle, as compared with a four-stroke one, increases the power-per-litre of the engine twofold, but in actual fact, it is better by only 40-70 per cent. This can be explained by the loss of part of the cylinder working volume due to the scavenging ports, expenditure of some power on compression of the scavenging air or mixture and in carburettor engines by the loss of part of the combustible mixture when combustion products are expelled from the cylinders.

A higher product $\eta_v \rho_0$ increases the charge mass and the power-per-litre of the engine, since in this case more fuel is burned in a unit of a cylinder working volume.

The admission coefficient η_v and the charge density ρ_0 can be increased by reducing the hydraulic resistances and the heating of the charge during admission and also with the aid of supercharging.

In up-to-date automobile engines with high compression ratios and side valves, the coefficient η_v is small, since the reduction in the dimensions of the combustion chamber tends to decrease the area through which the mixture is admitted into the cylinder. With mean compression ratios, the fuel intake area is sufficient, but when the compression ratio is increased this area is virtually reduced so much that it causes an appreciable drop in the coefficient η_v and, hence, in the mean effective pressure.

The overhead (suspension) valves used instead of the side ones make it possible to preserve or even increase the coefficient η_v not only when a higher compression ratio is used but also with a rise in the engine crankshaft angular velocity.

Figure 4.5 shows the growth (in per cent) of mean effective pressure *versus* the compression ratio with overhead and side valves. In the case of side valves (curve *1*), a rise in the com-

pression ratio from 6 to 9.5 increases the mean effective pressure by 6 per cent and upon further rise the pressure begins to drop. This occurs primarily because of the reduction of the admission coefficient. With overhead valves (curve *2*), an increase in the compression ratio from 6 to 9.5 increases the mean effective pressure by 15 per cent. A further rise is attended by a growth in the mean effective pressure.

Lately, to increase the coefficient η_V of high-speed engines, duplex and even quadruple carburettors have come into use.

The product $\eta_V \rho_0$ can be increased by injecting light fuel into the cylinders with forced ignition of the mixture or by supercharging (see Chapter VI).

The mechanical efficiency η_m can be increased by reducing the losses due to friction and for actuating the engine auxiliary mechanisms. The value of the mechanical losses is affected by the material of the piston, the quality of machining and assembly of the engine mated parts.

It seems from formula (4.29) that in all conditions the power-per-litre developed by the engine is directly proportional to the angular velocity of the engine crankshaft. However, this relationship is true only for a narrow range of angular velocities with a slight change in the pressure p_e or the torque M_e, i.e., near the maximum values of $p_{e\,\max}$ or $M_{e\,\max}$. A higher power-per-litre of an engine with increased angular velocity is possible only when the working process is effective, the value of the coefficient η_V is high and the mechanical efficiency reduced insignificantly.

Since the value of the $\frac{\eta_i}{\alpha}$ ratio which describes the course of the working process changes but slightly with an increase in the angular velocity, the change in the power-per-litre caused by the angular velocity primarily depends on the product $\eta_V \eta_m \omega_e$. Therefore, when the angular velocity of the engine crankshaft is increased, the maximum values of the coefficients η_V and η_m are to be obtained, which depend mainly on the hydraulic resistances at the intake and the mean speed of the piston.

The most effective methods of increasing the coefficient η_V are the installation of overhead valves and the reduction in the $\frac{S}{D}$ ratio, since this allows an increase in the valve intake areas and a shorter path to be traversed by the combustible mixture admitted into the cylinder.

A reduction in the $\frac{S}{D}$ ratio makes it possible not only to improve the charging of the engine but also to decrease the piston mean speed equal to $\frac{S\omega_e}{\pi}$.

It has been established experimentally that up to 60-65 per cent of all mechanical losses in the engine occur due to friction of the pistons and piston rings against the cylinder walls, whereas friction in the crankshaft bearings amounts to about 7 per cent of all losses. An increase in the piston mean speed diminishes mechanical efficiency, intensifies wear of the rubbing parts of the engine and increases the forces of inertia of the reciprocating masses. To balance these factors, the $\frac{S}{D}$ ratio is 0.8-1.0 in up-to-date high-speed automobile engines.

In addition to the above-listed advantages, a short-stroke design allows the specific mass of the engine to be reduced, i.e., the mass per kW or unit of working volume, the heat losses decreased, the stiffness of the crankshaft improved and the engine itself made more compact. The use of a short-stroke design with a *V*-type engine allows the length and mass of the engine to be reduced.

All this prompts the conclusion that a higher compression ratio, the use of overhead valves, increased angular velocity of the engine crankshaft and a short-stroke design allow a much higher power-per-litre to be obtained, the fuel consumption and the specific mass of the engines reduced.

Chapter V CHARACTERISTICS OF ENGINES

Motor vehicle engines can be operated over a large angular velocity range. The minimum angular velocity is restricted by conditions necessary for stable operation of the engine and the maximum one, by the efficiency of the gas exchange process, thermal stress of the parts, increased inertia loads, reduction in the mechanical efficiency, etc. Within this range of angular velocities the engine operates under different loads.

The optimum operating conditions of an engine are those which ensure, for example, either maximum power $N_{e\,max}$, or maximum torque $M_{e\,max}$, or minimum specific fuel consumption $g_{e\,min}$. In each particular case, these operating conditions are determined by the purpose of the engine and the specific features of its work.

The functioning of a motor vehicle engine is determined by the condition of equality of the effective power of the engine and resistances to motion (power consumed which usually changes within a rather broad range).

The power consumed depends not only on the speed of a motor vehicle but, to a large extent, also on such factors as the state of the road, the load, the intensity of acceleration, etc. The operating conditions of an engine are unsteady, since the load on the engine continually changes. In case of a short-time increase in the resistance to motion, the latter can be overcome at the expense of the kinetic energy accumulated by the motor vehicle in acceleration, and with more prolonged and high resistance, at the expense of the store of the engine torque. The torque in carburettor engines is increased by opening the throttle, and in diesel engines by turning the plungers of the high-pressure fuel pump towards an increase in the fuel delivery. For this reason, engine operation analysis should not be confined to the study of only some one constant duty, but research should be conducted into engine operation at various duties.

The power and economy of engine operation in various conditions are evaluated from the engine characteristics. *The engine characteristic* is a graphical representation of the dependence of such basic parameters of its operation as power, torque, fuel consumption on angular velocity, load and other engine parameters.

The principal characteristics of motor vehicle engines are *speed, load and adjustment* characteristics.

The power and economy of engine operation depend essentially on the adjustment of its units. Therefore, before the basic characteristics of an engine are taken, it is necessary to take its adjustment characteristics. The latter are required to estimate the optimum indices of engine operation depending on the composition of the combustible mixture, the ignition advance angle, the amount of injected fuel, the moment of its delivery, etc.

5.1. Speed Characteristics

The speed characteristic of an engine is a graphical representation of the dependence of the basic parameters describing the functioning of the engine (power, torque, etc.) on the crankshaft

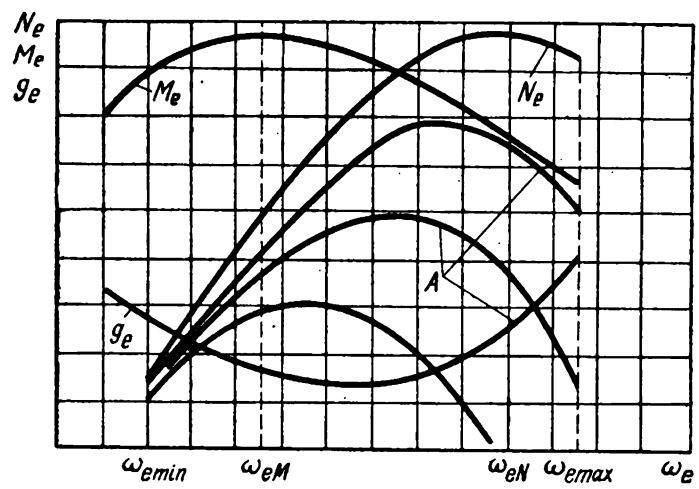

Fig. 5.1. External and part-load (throttle) speed characteristics of a carburettor engine

angular velocity with the throttle or the fuel pump rack in constant position and also with the constant values of the temperature of the oil and cooling water. The speed characteristic graphs are constructed with the positions of the throttle and pump rack constant but different for each characteristic.

The speed characteristic which corresponds to the wide-open throttle or the maximum delivery of fuel is known as *external* (curves N_e and M_e in Fig. 5.1). Thus, the external speed characteristic is used to determine the maximum power that can be developed by a given engine with various angular velocities. The characteristics obtained with a partly opened throttle or with partial delivery of fuel are commonly referred to as the *part-load speed characteristics* (curves *A* in Fig. 5.1).

Distinction is made between the following typical angular velocities of the engine crankshaft:

ω_{min} = minimum angular velocity at which the engine operates stably under full load; in motor vehicle engines the angular velocity ω_{min} is within 40-60 rad/s; this speed diminishes at idle run

ω_M = anguar velocity corresponding to the maximum values of torque $M_{e\,max}$ and mean effective pressure p_e

ω_N = angular velocity corresponding to the maximum engine power

ω_{max} = angular velocity corresponding to the maximum motor vehicle speed (when a limiter or a regulator is installed, this speed is equal to the angular velocity at which it begins to operate)

In actual conditions a motor vehicle carburettor engine mainly operates within the interval of angular velocities ω_M — ω_N. Within this interval the transmission gears are usually changed over and the minimum specific fuel consumption $g_{e\,min}$ is estimated. The best fuel saving of an engine can be ensured with the minimum changes in the specific fuel consumption within this interval of angular velocities.

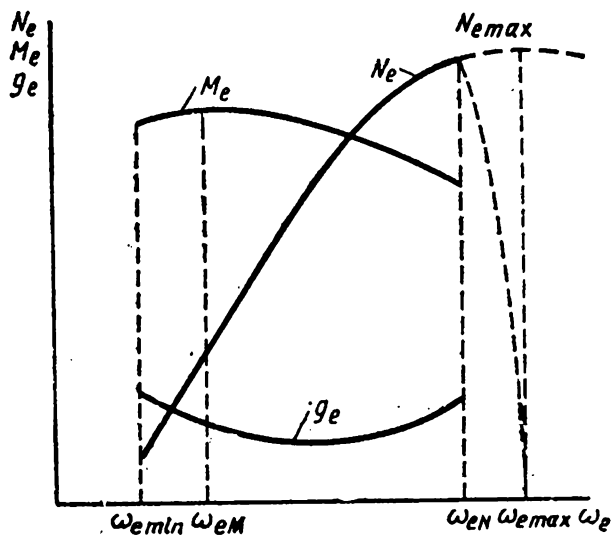

Fig. 5.2. Speed characteristic of a diesel engine operating with an angular velocity regulator

The speed characteristic of a diesel engine operating with an angular velocity regulator is illustrated in Fig. 5.2. The maximum power N_{emax} of a diesel engine lies in the smoking zone and is arranged beyond the regulated branch of the curve.

It follows from the above speed characteristics that, as the angular velocity grows, the effective power of the engine first increases and then diminishes upon reaching its maximum value. The engine develops the maximum power $N_{e\,max}$ at the moment when the increase in power with a growth in angular velocity (frequency of cycles) is completely compensated by the reduction in the mean effective pressure p_e. At a higher angular velocity the pressure p_e is reduced because the charging of the engine deteriorates and the mechanical losses are increased.

The speed characteristic of an actual (produced) engine is plotted on the basis of the data obtained from tests on a special stand. If the speed characteristic is to be constructed for a newly

designed engine, it is plotted from the results of the thermal calculations or with the aid of empirical formulas.

To plot the speed characteristic from thermal heat balance is difficult, as the lack of experimental data for various speed duties of engines of different types makes it impossible to specify the initial values with sufficient accuracy. Therefore, this method of constructing the speed characteristic of an engine does not always yield satisfactory results and this characteristic for a newly designed engine is commonly plotted with the aid of one of the empirical formulas.

The speed characteristic of an engine can also be constructed from an empirical formula if the maximum power $N_{e\,max}$ (in kW) and the angular velocity ω_N (in rad/s) are known.

Nowadays several empirical formulas suggested by various authors are employed. Some of them are illustrated below.

For carburettor engines

$$N_e = N_{e\,max}\left[\frac{\omega_e}{\omega_N} + \left(\frac{\omega_e}{\omega_N}\right)^2 - \left(\frac{\omega_e}{\omega_N}\right)^3\right] \tag{5.1}$$

for diesel engines with an open combustion chamber

$$N_e = N_{e\,max}\left[0.87\frac{\omega_e}{\omega_N} + 1.13\left(\frac{\omega_e}{\omega_N}\right)^2 - \left(\frac{\omega_e}{\omega_N}\right)^3\right] \tag{5.2}$$

for diesel engines with a prechamber

$$N_e = N_{e\,max}\left[0.6\frac{\omega_e}{\omega_N} + 1.4\left(\frac{\omega_e}{\omega_N}\right)^2 - \left(\frac{\omega_e}{\omega_N}\right)^3\right] \tag{5.3}$$

for diesel engines with a vortex combustion chamber

$$N_e = N_{e\,max}\left[0.7\frac{\omega_e}{\omega_N} + 1.3\left(\frac{\omega_e}{\omega_N}\right)^2 - \left(\frac{\omega_e}{\omega_N}\right)^3\right] \tag{5.4}$$

where N_e and ω_e are the effective power in kW and angular velocity of the crankshaft in rad/s at the sought-for point of the engine speed characteristic.

Having specified the angular velocities ω_e and estimated their corresponding values of the power N_e, the speed characteristic of the power $N_e = f(\omega_e)$ can be plotted and then the speed characteristic of the torque $M_e = f(\omega_e)$. The values of the torque M_e can be found from the formula:

$$M_e = \frac{N_e}{\omega_e}\,10^3 \tag{5.5}$$

Example. Calculate the external speed characteristic of a four-stroke carburettor engine developing a maximum effective power of $N_{e\,max} = 72$ kW with a crankshaft angular velocity of $\omega_N = 472$ rad/s.

The effective power with an angular velocity of 136 rad/s [formula (5.1)]

$$N_e = 72\left[\frac{136}{472} + \left(\frac{136}{472}\right)^2 - \left(\frac{136}{472}\right)^3\right] = 25.0 \text{ kW}$$

The effective torque at the same angular velocity [formula (5.5)]

$$M_e = \frac{25{,}000}{136} = 184 \text{ N} \cdot \text{m}$$

The values of power and torque with other values of crankshaft angular velocity are illustrated in the table below.

Parameters	Angular velocity of crankshaft in rad/s					
	136	220	304	388	472	556
N_e in kW	25.0	41.9	57.0	67.6	72.0	67.5
M_e in N · m	184	190	188	174	152	121

The external speed characteristics of the carburettor engines ГАЗ-51 and ЗИЛ-130 are shown in Fig. 5.3 and those of the diesel engines ЯМЗ-236 and ЯМЗ-238 in Fig. 5.4.

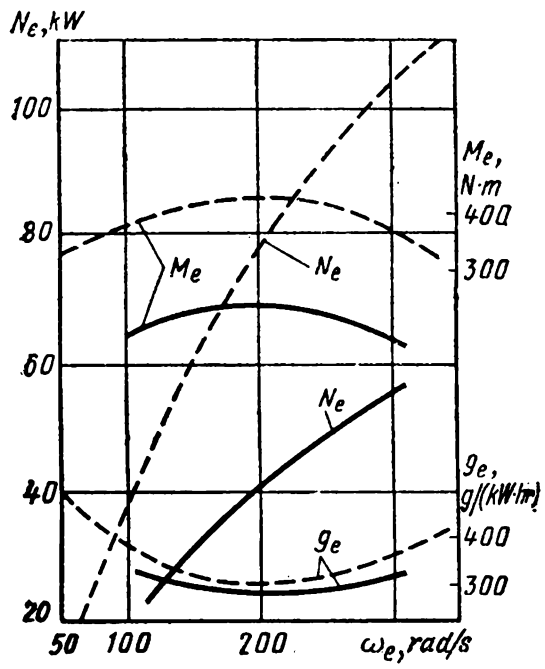

Fig. 5.3. External speed characteristics of carburettor engines
solid lines—ГАЗ-51 engine; dash lines—ЗИЛ-130 engine

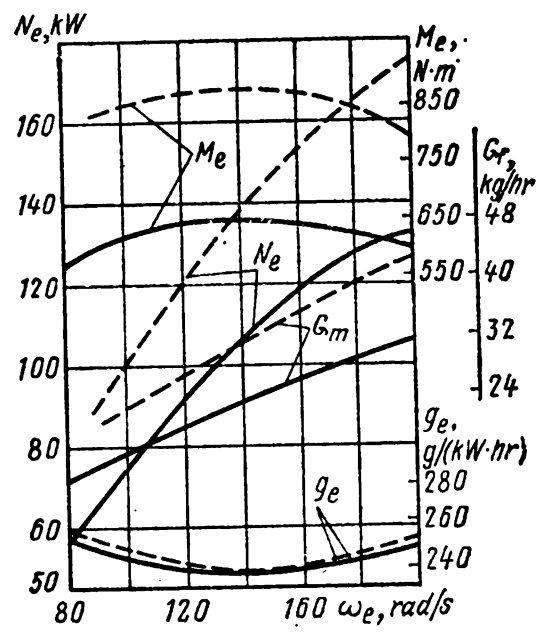

Fig. 5.4. External speed characteristics of diesel engines
solid lines—ЯМЗ-236 engine; dash lines—ЯМЗ-238 engine

It can be seen from the above speed characteristics that the maximum values of torques $M_{e\,\max}$ are displaced with respect to the maximum values of power $N_{e\,\max}$ to the zone of smaller angular velocities. The values of the ratio of the angular velocity ω_M with the maximum torque to the angular velocity ω_N with the maximum power are within 0.4-0.7 in up-to-date automobile engines.

An increase in the torque to the $M_{e\,\max}$ value with the angular velocity reduced from ω_N to ω_M influences greatly the stable

operation of a motor vehicle engine. The stable duty is estimated by the torque reserve called the *adaptability coefficient* k_M which is the ratio of the maximum torque $M_{e\,max}$ to the torque at maximum power:

$$k_M = \frac{M_{e\,max}}{M_{e\,N}}$$

The adaptability coefficient describes the property of an engine to overcome short-time increased loads.

The values of the adaptability coefficient are given below:

Carburettor engines	1.25-1.35
Diesel engines	1.05-1.15

A graph showing the dependence of the specific fuel consumption g_e on angular velocity ω_e with a wide-open throttle or with full delivery of fuel is plotted for the engine fuel economy to be estimated.

The specific fuel consumption g_e [in g/(kW·hr)] is determined from the engine thermal calculations or from the formula:

$$g_e = 1000 \frac{G_f}{N_e} \tag{5.6}$$

where G_f is the hourly fuel consumption found experimentally in kg/hr.

5.2. Load Characteristics

Motor vehicle engines operate not only at full throttle or with a full delivery of fuel but also at other duties. For this reason, the speed characteristic alone cannot estimate in full the properties of an engine in various operating conditions.

In order to determine the fuel economy of an engine under different loads, *load characteristics* are taken. The load on a carburettor engine is altered by changing the position of the throttle. Therefore, its load characteristic is also known as the *throttle characteristic.*

The load characteristic is the graphically expressed dependence of the specific fuel consumption g_e on the effective power N_e (or mean effective pressure p_e) at a constant angular velocity of the crankshaft. Since a motor vehicle engine operates within a very broad range of angular velocities, it is usual to take not one but several load characteristics.

Figure 5.5 shows the load characteristic of a carburettor engine. The values of load (power) are plotted on the X-axis in per cent of the power developed by the engine with the wide-open throttle and with a given angular velocity. The values of the hourly and specific fuel consumption are plotted on the Y-axis.

When the throttle is opened wide the specific fuel consumption is equal to the consumption determined by the speed characteristic at the same angular velocity. At idle run, when the power $N_e = 0$ and the hourly consumption of fuel has a finite value, the specific fuel consumption is equal to infinity.

Each curve of the load characteristic corresponds to one constant angular velocity, transition from one point of the curve to another being effected by changing the degree by which the throttle is opened, while the constant angular velocity is mainained by altering the load on the engine with the aid of a hydraulic or electric brake.

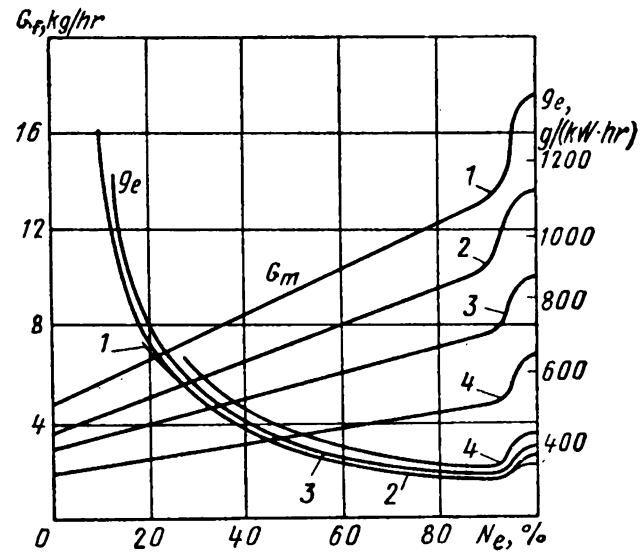

Fig. 5.5. Load characteristic of a carburettor engine
1—ω_e=189 rad/s, *2*—ω_e=147 rad/s, *3*—ω_e = =105 rad/s, *4*—ω_e=63 rad/s

The hourly fuel consumption changes almost by a linear law. The sharp upward bends of the curves G_f and g_e with loads close to the maximum are caused by the considerable combustible mixture enrichment by the carburettor economizer.

An increase in the specific fuel consumption with the throttle slightly closed occurs due to the impaired working process of the engine (worse charging and increased coefficient of residual gases) and also reduced mechanical efficiency (the engine power drops and the amount of mechanical losses practically remains constant at a constant angular velocity).

5.3. Adjustment Characteristics

The curves showing the dependence of engine power and economy on fuel consumption, mixture composition, temperature of oil and water, timing angle, etc., are known as *adjustment characteristics*. These characteristics are necessary to reveal the most advantageous engine operation conditions depending on the above factors and to estimate the adjustment quality.

The adjustment characteristics are taken with full and partial loads. Most often the adjustment characteristics are taken with respect to fuel consumption; they show the change in power N_e and specific fuel consumption g_e *versus* hourly consumption G_f at a constant angular velocity of the crankshaft and an optimum angle of ignition advance.

The adjustment characteristic as to fuel consumption of a carburettor engine with $\omega_e = 210$ rad/s is illustrated in Fig. 5.6. It has two distinctive points, one corresponding to maximum power and the other, to minimum specific fuel consumption.

The engine develops maximum power when its hourly fuel consumption corresponds to a slightly enriched mixture ($\alpha = 0.8$-0.9),

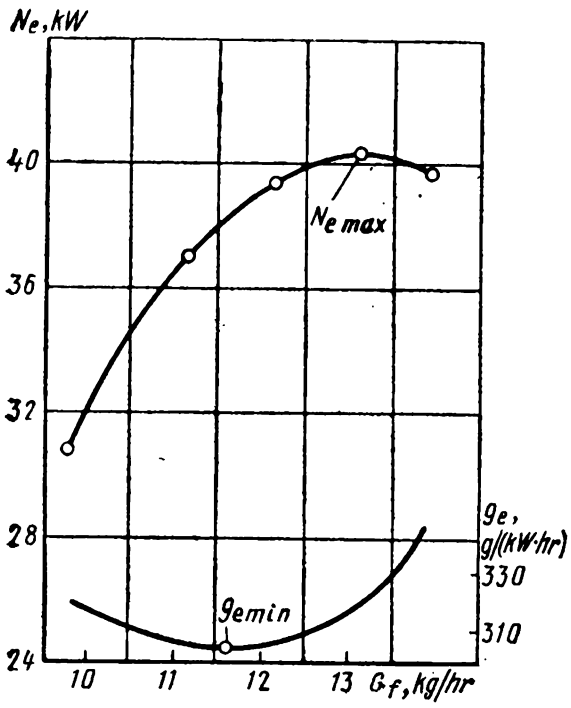

Fig. 5.6. Adjustment characteristic of a carburettor engine as to fuel consumption G_f ($\omega_e = 210$ rad/s)

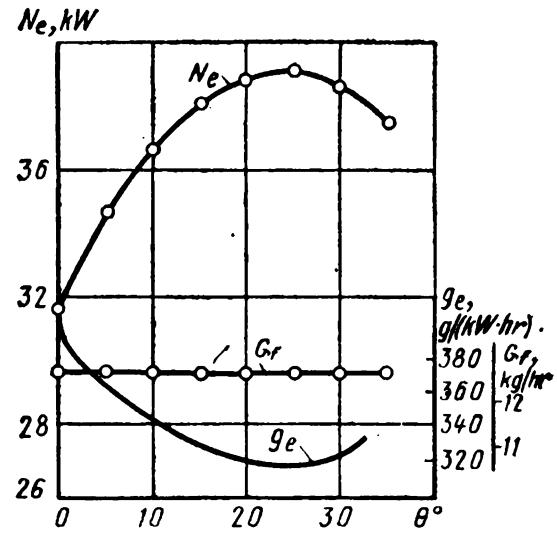

Fig. 5.7. Adjustment characteristic of a carburettor engine as to ignition advance angle θ ($\omega_e = 210$ rad/s)

since in this case the rate of combustion increases. When the mixture is weakened the power drops because the combustion rate and the heat of mixture combustion are reduced.

The best fuel saving of an engine is obtained when the hourly consumption of fuel corresponds to a somewhat leaner adjustment ($\alpha = 1.1$-1.2). Further leaning of the combustible mixture causes an appreciable drop in the combustion rate and makes the engine function unsteadily which is attended by a sharp drop in power and poor fuel economy.

Thus, the region of operational adjustments of a carburettor should lie between the minimum fuel consumption and the maximum engine power. It should also be recalled that the operation of an engine beyond the limits of these duties is undesirable, because it is accompanied by a simultaneous reduction of power and deterioration of engine fuel economy.

In practice, the carburettor is never adjusted to the maximum power at full throttle; this power is normally reduced by 1-2 per cent to improve fuel economy.

The adjustment characteristic as to the ignition advance angle at an angular velocity of $\omega_e = 210$ rad/s is shown in Fig. 5.7.

This characteristic shows that with an increase in the ignition advance angle θ to 25°, the power of the engine increases and the specific fuel consumption diminishes. Upon further growth in the angle θ, the power of the engine drops and the specific fuel consumption increases. Therefore, the optimum ignition advance angle at this duty is equal to 25°.

5.4. Detonation Characteristics

The tendency of fuel to knock determines the design features of an engine. The anti-knock properties of an engine are estimated from the detonation characteristics.

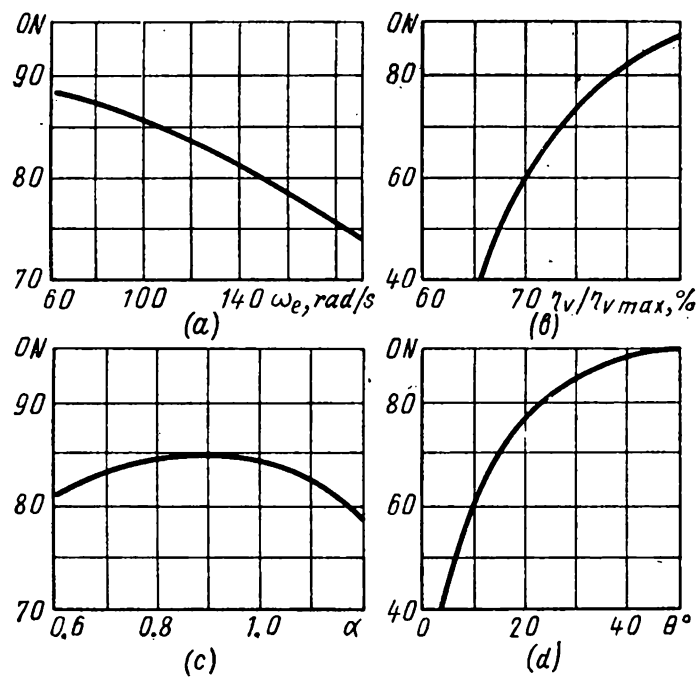

Fig. 5.8. Necessary fuel octane number ON versus the basic duty parameters of an engine ω_e, $\eta/\eta_{V\,max}$, α and θ

The engine detonation characteristic is a graphically represented dependence of the fuel octane number necessary for engine operation at the boundary of detonation for any one particular engine duty parameter.

Figure 5.8 illustrates the detonation characteristics, i.e., graphs of dependence of petrol octane numbers on such duty parameters of the engine as ω_e, $\eta_V/\eta_{V\,max}$, α and θ. The characteristics are plotted for one and the same engine.

The graph of dependence of the fuel octane number on the crankshaft angular velocity (Fig. 5.8*a*) is constructed for a full-open throttle, carburettor power adjusted, and ignition advance

angle being optimum. The graph shows that the fuel octane number should be higher at small angular velocities.

The graph of fuel octane number *versus* engine load (the ratio $\eta_V/\eta_{V\,max}$, Fig. 5.8*b*) is plotted for a constant angular velocity, carburettor power adjusted and optimum angle of ignition advance. It follows from the graph that the maximum fuel octane number occurs with the wide-open throttle. When the engine is throttled ($\eta_V/\eta_{V\,max} < 1$) the required value of the octane number diminishes.

The graph of dependence of the fuel octane number on the mixture composition (Fig. 5.8*c*) has been obtained with a full-open throttle, constant angular velocity and optimum angle of ignition advance. Octane numbers should be higher with a rich mixture.

The graph of dependence of the fuel octane number on the ignition advance angle (Fig. 5.8*d*) has been obtained with a full-open throttle, constant angular velocity and power adjustment of the carburettor. With an increase in ignition advance angle the fuel octane number should be increased.

Chapter VI FUTURE DEVELOPMENTS OF MOTOR VEHICLE ENGINES

The further development and perfection in the design of motor vehicle engines goes along the lines of increasing the power-per-litre, improving the fuel economy, reducing the overall dimensions and the unit mass, and prolonging the service life.

The most promising engines are those with supercharging, turbocharging, direct injection of light fuel and forced ignition, fuel spray ignition as well as rotor-piston and multi-fuel engines.

6.1. Automobile Gas-Turbine Installations

In the last twenty years, due to their advantages over the internal-combustion reciprocating engines, motor vehicle gas-turbine engines have been developed.

The potential energy of the combustion products is utilized in a reciprocating engine to effect an expansion stroke by the piston, it is converted in a gas turbine into the gas flux kinetic energy which is imparted to the turbine rotor.

Two-Shaft Gas-Turbine Installation

The principal diagram of an automobile two-shaft gas-turbine installation with heat regeneration is illustrated in Fig. 6.1.

The installation operates as follows. The air sucked in from the atmosphere through pipe *3* is compressed in centrifugal compressor *4*, passes via regenerator (heat exchanger) *6* and enters combustion chamber *11* along pipe conduit *7*. Fuel pump *1* delivers fuel along fuel pipe *16* to atomizer *10*, through which it is injected into the combustion chamber. The products of combustion flow from combustion chamber *11* via guide blades *15* and rotating blades *5* of compressor turbine *14* and then through guide blades *13* and rotating blades *12* of power turbine *8* to heat exchanger *6*. The compressor turbine actuates the compressor and the auxiliary mechanisms (fuel *1* and oil *2* pumps, electric generator). Transmission *9* connects the power turbine with the driving wheels of the motor vehicle. The combustion products pass through an annular duct *A* and heat exchanger *6* in which they give off part

of the heat to the air entering combustion chamber *11* along pipe conduit *7*.

The air in the compressor is compressed to 0.40-0.45 MN/m^2, the peripheral velocity of the blades reaching 400-500 m/s and the angular velocity 3-4 ths rad/s.

The fixed guide blades (nozzle box assemblies) *13* and *15* of the turbines are arranged circumferentially and form ducts through which the gases pass at a rate of 500-700 m/s. The gases then

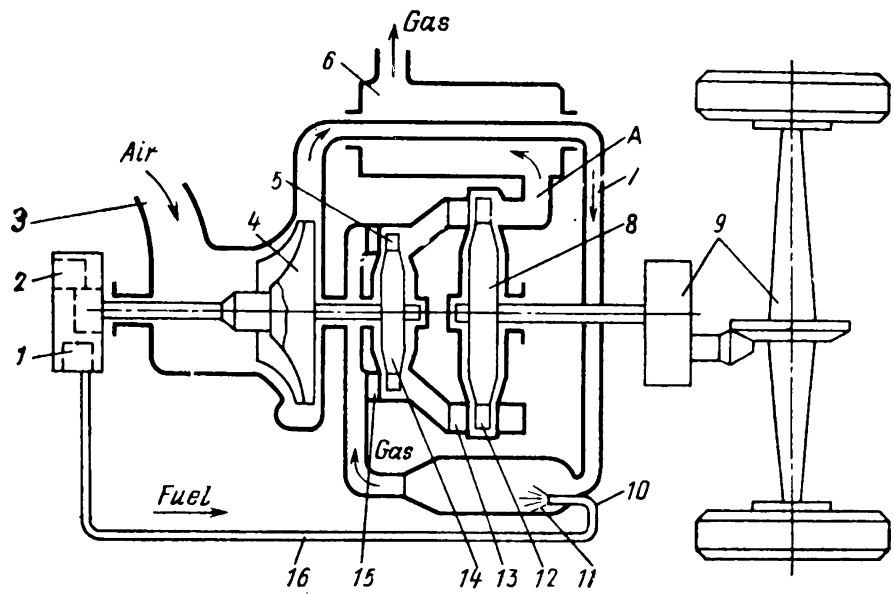

Fig. 6.1. Automobile gas-turbine installation

A—duct; *1*—fuel pump; *2*—oil pump; *3*—pipe; *4*—centrifugal compressor; *5*—rotating blades of compressor turbine; *6*—heat exchanger; *7*—pipe conduit; *8*—power turbine; *9*—transmission; *10*—atomizer; *11*—combustion chamber; *12*—rotating blades of power turbine; *13*—guide blade of pulling turbine; *14*—compressor turbine; *15*—guide blade of compressor turbine; *16*—fuel pipe

strike rotating blades *5* and *12* which rotate the disks and shafts of turbines *8* and *14*.

This type of an automobile gas-turbine installation is characterized by the fact that both fuel and air are admitted into the combustion chamber continually, and the combustible mixture is burned in an open chamber at a constant pressure.

For motor vehicles use is mainly made of two-shaft gas-turbine installations in which the disks of the compressor and power turbines are mounted on separate shafts. In this case the turbine disks may develop various angular velocities: the compressor turbine — a higher angular velocity to ensure high efficiency of the centrifugal compressor, and the power turbine — a small angular velocity during operation or in starting.

In a two-shaft installation, the decrease in the angular velocity of the power turbine increases the torque and at the minimum an-

gular velocity the torque exceeds 2-2.5 times the value at the nominal duty. Besides, the two-shaft design ensures better fuel saving than a single-shaft design under partial loads.

Motor vehicle gas-turbine engines have the following main advantages over reciprocating engines:

1. Smaller overall dimensions and mass.
2. Better balance.
3. The use of inexpensive liquid fuel.
4. Better tractive characteristic, because the torque increases as the angular velocity drops.
5. Easy starting at various temperatures and absence of a cooling system.

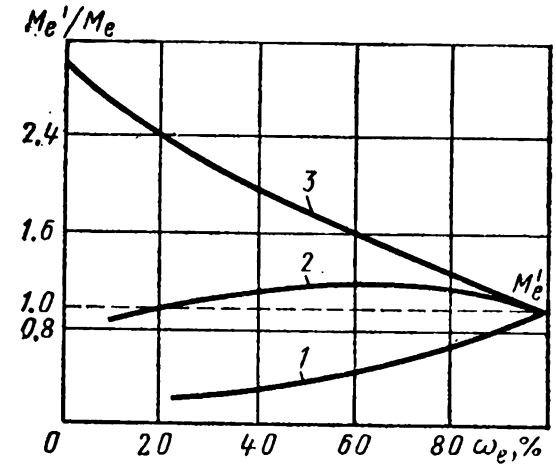

Fig. 6.2. Torque versus angular velocity of the shaft

1—single-shaft gas-turbine installation; *2*—internal-combustion reciprocating engine; *3*—two-shaft gas-turbine installation

Alongside their advantages, modern automobile gas-turbine engines have pronounced disadvantages. At the optimum duty and under partial loads their fuel economy is low and the brake thermal efficiency is within 12-16 per cent.

The principal causes of their worse fuel economy are the reduced temperature of the gases coming to the turbine blades and also the small effective power they develop. The temperature of the working gases before the nozzle box assembly of the turbine is restricted by the quality of the heat-resistant steel, which tends to reduce the brake thermal efficiency of the gas-turbine engine. A high excess of air ($\alpha = 3.5$-4.5) and, therefore, a high capacity of the compressor is needed to reduce the temperature of the gases. As a result, about 60-70 per cent of the power of the gas-turbine engine is expended on driving the compressor. The power of the compressor turbine in a two-shaft installation should be 1.5-2 times larger than the power of the power turbine.

The maximum power of commercial engines does not exceed 450 kW, i.e., it is rather low for a gas turbine. For this reason, the size of the blades is small, which causes rather high losses in the turbine blading and reduces the efficiency of the turbine and the compressor.

The fuel economy of a gas-turbine installation can be improved by increasing the angular velocity of the turbine shaft, the compression ratio of air in the compressor and the temperature of the working gas before the nozzle box assembly, and by using a regenerator.

The Fords have designed a three-shaft gas-turbine installation rated at 220 kW and operating on a complicated cycle with intermediate air cooling, regeneration of heat and two-stage combustion. Despite the cycle complexity, the unit mass of such an installiton is 1.4 kg/kW. The specific fuel consumption at a temperature of gas before the turbine of 1200 K and the maximum power is 345 g/(kW·hr). The total pressure increase ratio π_K in two stages of the centrifugal compressor is 16. The angular velocity of a low-pressure compressor is 4750 and that of a high-pressure compressor, 9550 rad/s. The shaft of the power turbine rotates at 3750 rad/s. The high-pressure compressor is driven by a centripetal turbine. The power turbine and the turbine that drives the low-pressure compressor are of the axial-flow type.

Figure 6.2 shows the torques of a single-shaft gas-turbine installation (curve *1*), an internal-combustion reciprocating engine (curve *2*) and a two-shaft gas-turbine installation (curve *3*) *versus* angular velocity ω_e per cent. It can be seen on the graph that in its tractive properties a two-shaft gas-turbine installation is considerably superior to an internal-combustion reciprocating engine, a very important advantage for a transport power unit.

6.2. Supercharging of Engines

Engines are supercharged by centrifugal or volume displacement superchargers.

A centrifugal supercharger is actuated by the crankshaft of the engine (Fig. 6.3*a*) or by a special gas turbine utilizing the energy of the exhaust gases (Fig. 6.3*b*). In the latter case the engine fuel economy is improved due to the use of kinetic energy of the exhaust gases.

A supercharger driven by the crankshaft operates as follows. Atmospheric air enters supercharger *1* (Fig. 6.3*a*) where it is compressed to the required pressure. During the admission stroke the compressed air flows along pipe conduit *2* into the engine cylinder. In this case, some of the power developed by the engine is used to drive the supercharger.

A gas-turbine supercharger operates in a slightly different manner. The used gases of the engine flow along pipe conduit *4* (Fig. 6.3*b*) and through nozzle box assembly *5* to blades *8* of the gas turbine and perform useful mechanical work after which they are withdrawn along pipe conduit *6*. The wheel of the gas turbine and the impeller of supercharger *7* are mounted on one common shaft. The air compressed by the supercharger through pipe conduit *3* enters the cylinder which appreciably increases the engine admission coefficient.

A mass charging of the cylinder with air in supercharging is higher. For this reason, much more fuel can be burned in the cylinder thus ensuring higher temperatures and pressures. The supercharging pressure in diesel engines is restricted by thermal and mechanical stress of parts in the crank mechanism. The air pressure in carburettor engines mainly limits the detonating combustion of fuel, and the compression ratio should be reduced (to a definite limit) when supercharging.

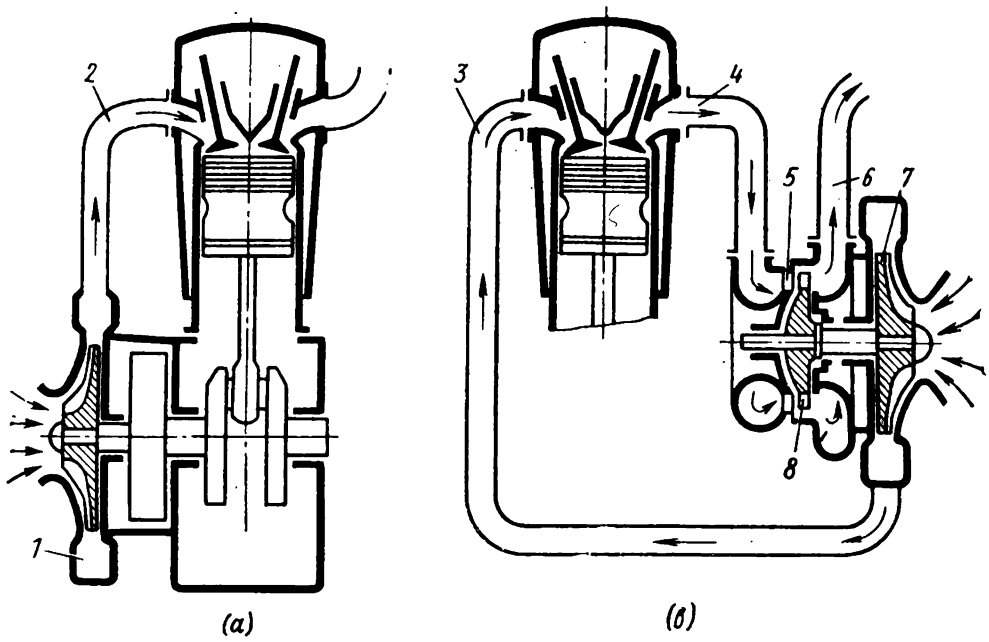

Fig. 6.3. Centrifugal supercharger drives for supercharging an engine

(*a*) mechanical drive; (*b*) gas-turbine drive; *1*—supercharger; *2*, *3*, *4* and *6*—pipe conduits; *5*—nozzle box assembly; *7*—impeller; *8*—gas turbine blades

The supercharger of a carburettor engine is mounted before the carburettor (Fig. 6.4*a*) or behind it (Fig. 6.4*b*). In the first instance, the air is compressed in the supercharger and in the second, the combustible mixture.

If the supercharger is installed before the carburettor the latter is under pressure and it should be sealed sufficiently. In this case, it is possible to reduce the carburettor size and dispense with mixture heating after starting and during the engine warming period at small angular velocities, because the temperature of the air issuing from the supercharger is high enough.

If the supercharger is arranged behind the carburettor, a large space filled with combustible mixture is formed between the carburettor and the cylinder. This may cause high fuel condensation when starting and warming up the engine.

Volume displacement superchargers can be designed with one (Fig. 6.4*c*) and two (Fig. 6.4*d*) rotors. The operating principle of a single-rotor supercharger (see Fig. 6.4*c*) is as follows. Shaft *1*

mounted eccentrically in drum *4* of housing *5* carries rotor *6* with radial slots *A* into which blades *3* are inserted. As the rotor rotates, the blades acted upon by the centrifugal forces are pressed against drum *4* with ports *2*, as a result the space between the neighbouring blades diminishes and the air in it is compressed. The drum is freely mounted in the housing and can rotate to suit blade wear.

The principle of action of a double-rotor supercharger (see Fig. 6.4*d*) is as follows. Two double-blade rotors *6* are mounted on two parallel shafts *1* arranged in housing *5* of the supercharger.

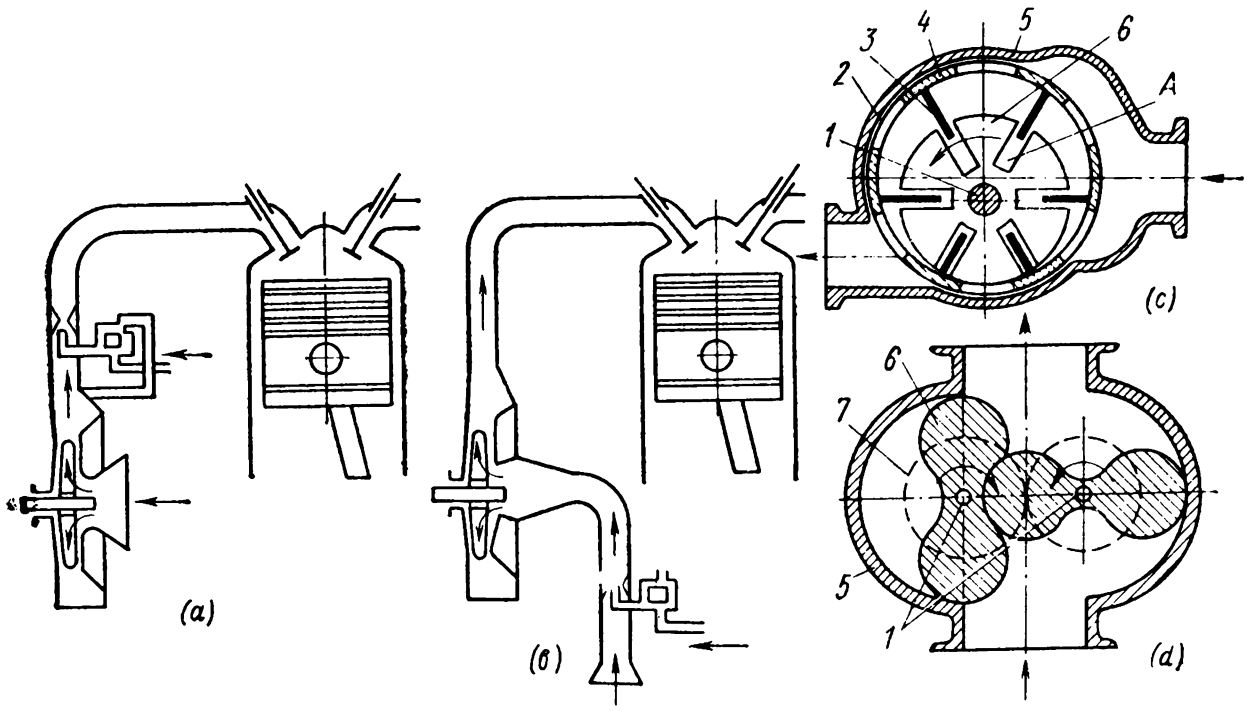

Fig. 6.4. Arrangement of centrifugal supercharger and diagrams of volume displacement superchargers

(*a*) before carburettor; (*b*) behind carburettor; (*c*) single-rotor supercharger; (*d*) double-rotor supercharger; *A*—rotor slots; *1*—shafts; *2*—port; *3*—blades; *4*—drum; *5*—housing; *6*—rotor; *7*—gear

The shafts of the rotors are linked by gears *7* and rotated in opposite directions. Acted upon by rarefaction, the air enters the supercharger through the bottom pipe and fills the space between the rotors and the housing. Then, this amount of air is transferred into the delivery end and is expelled through the upper pipe.

Superchargers are rarely employed in carburettor engines, since they noticeably increase the thermal and mechanical stress of the crank mechanism parts and thus reduce the engine service life. In order to diminish the wear of a motor vehicle engine in supercharging, the supercharger should be engaged only under full load and disengaged in the case of partial loads.

Supercharging is very popular in the carburettor engines of racing cars. Superchargers for carburettor and diesel engines in-

stalled in motor vehicles operating in mountain country, when the power of engines may drop quite appreciably, give advantages.

The supercharger in two-stroke diesel engines is utilized not only for a slight blowing of air into the cylinders but also for scavenging purposes.

6.3. Injection of Light Fuel

Injection of light fuel into the cylinders of an engine with forced ignition of the mixture by an electric spark increases the admission coefficient due to the drop of the resistances in the

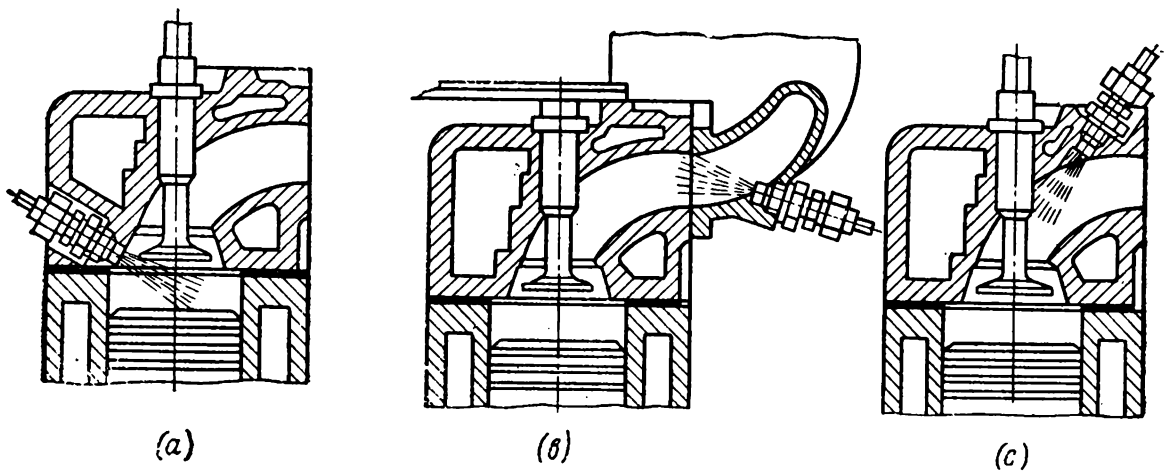

Fig. 6.5. Methods of fuel injection in engines with forced ignition
(*a*) into cylinder; (*b*) into intake pipe; (*c*) into intake duct

admission system, and the fuel is uniformly distributed among the cylinders.

The fuel is injected with the aid of a pump through an atomizer directly into the cylinder (Fig. 6.5*a*), or into the intake pipe (Fig. 6.5*b*) or the intake duct in the cylinder head (Fig. 6.5*c*).

When fuel is injected directly into the cylinder, it should be delivered in strictly measured portions. During injection into the intake pipe or the intake duct of the head the fuel should be delivered in portions and in a continuous stream. In this case, the admission coefficient increases because the resistances in the intake system are less (no carburettor and a larger area in the intake pipe, no preheating of the air admitted into the cylinders).

Uniform distribution of fuel among the engine cylinders creates favourable conditions for weak-mixture engine operation. Carburettor engines operate on mixtures with a lower excess air coefficient, since the composition of the mixture in separate cylinders may differ in this case by 10-20 per cent. Therefore, the injection of light fuel increases power and improves fuel economy. This is

especially important in two-stroke engines in which there is no loss of the combustible mixture during scavenging; this loss usually comes to 30 per cent and more.

The graph in Fig. 6.6 shows the dependence of mean effective pressure on the excess air coefficient for a single-cylinder engine with a carburettor (curve *1*) and injection of light fuel into the cylinder (curve *2*). It can be seen on the graph that the maximum rise of pressure p_e in a carburettor engine where petrol is injected at a pressure of p_e occurs when the engine operates with enriched mixtures.

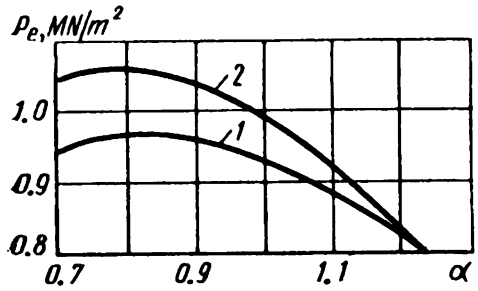

Fig. 6.6. Mean effective pressure p_e versus excess air coeficient α

6.4. Fuel Spray Ignition

Fuel spray ignition of the mixture ensures normal engine operation on lean mixtures, and improves the engine fuel economy.

The diagram of an engine combustion chamber with fuel spray ignition of the mixture is shown in Fig. 6.7. The combustion chamber consists of main chamber *1* and prechamber *8*. The combustible mixture enters the combustion chambers from carburettor *4* along pipe conduits *3* and *5*, the main chamber receiving a lean mixture and the prechamber, an enriched one.

The chambers are charged with lean and rich mixtures during admission through valves *2* and *6*. At the end of the compression stroke a spark between the electrodes of ignition plug *7* ignites the enriched mixture in the prechamber. The pressure in the prechamber sharply increases and, as a result, fuel sprays are ejected into the main chamber from the prechamber through the ducts *A*. Due to a large area of contact and an intensive vortex motion the flames ignite the lean mixture in the main chamber and make it burn at high combustion rates.

The use of fuel spray ignition of lean combustible mixture considerably improves the fuel saving. This method of ignition ensures the engine stable operation at the coefficient $\alpha = 1.4\text{-}2$. Besides, fuel spray ignition assures complete fuel combustion and allows a certain increase in the engine compression ratio.

Figure 6.8 illustrates the head of one of the cylinders of ГАЗ-51Ф engine with fuel spray ignition designed at the Gorky automobile works. The cylinder head has prechambers *2* connected with the main combustion chambers *6* by two ducts *A* arranged at an angle. The combustible mixture enters the chambers through

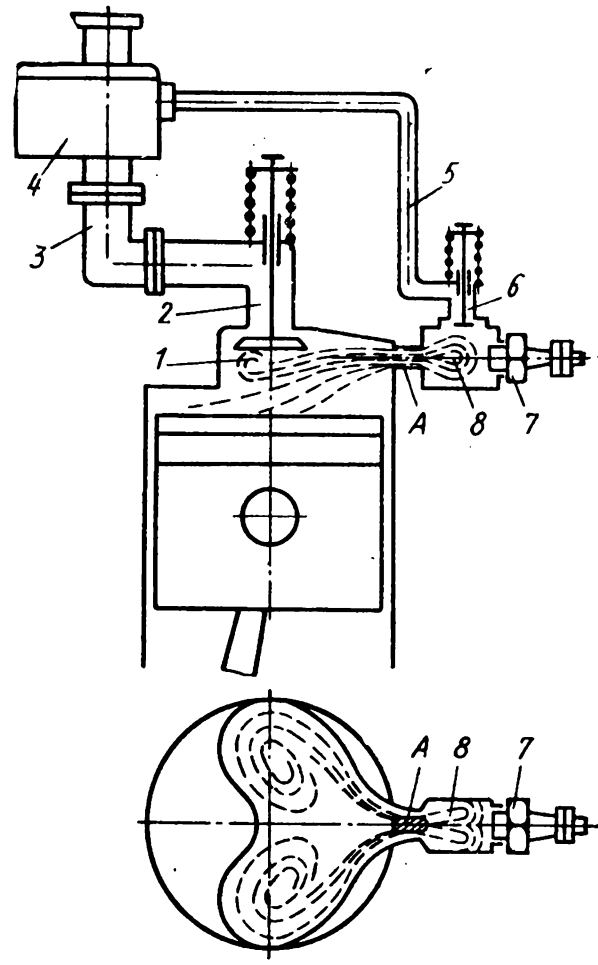

Fig. 6.7. Engine combustion chamber with fuel spray ignition of mixture

A—duct; *1*—main chamber; *2* and *6*—intake valves; *3* and *5*—pipe conduits; *4*—carburettor; *7*—ignition plug; *8*—prechamber

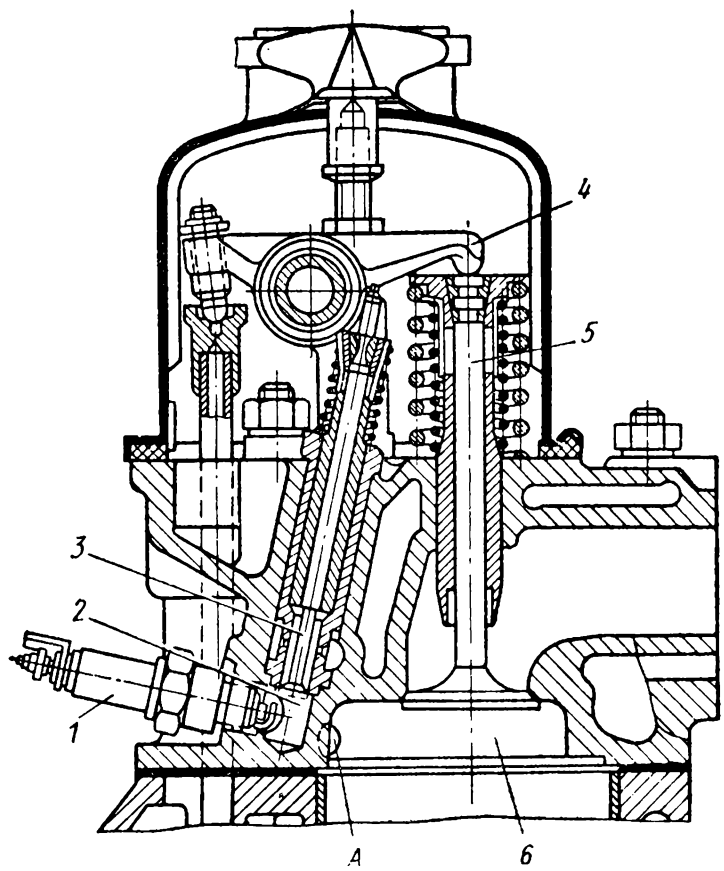

Fig. 6.8. Cylinder head of ГАЗ-51Ф engine with fuel spray ignition

A—duct; *1*—spark plug; *2*—prechamber; *3* and *5*—valves; *4*—rocker arm of valves; *6*—main chamber

valves *3* and *5* which are opened simultaneously by rocker arms *4*. Spark plug *1* is mounted in each prechamber.

The specific rates of fuel consumption for a ГАЗ-51 engine with side valves (curve *1* in Fig. 6.9) and for a ГАЗ-51Ф engine with overhead valves and fuel spray ignition (curve *2*) are compared with respect to the load characteristics of these engine. It follows from these curves that the use of fuel spray ignition saves much fuel especially under small and medium loads.

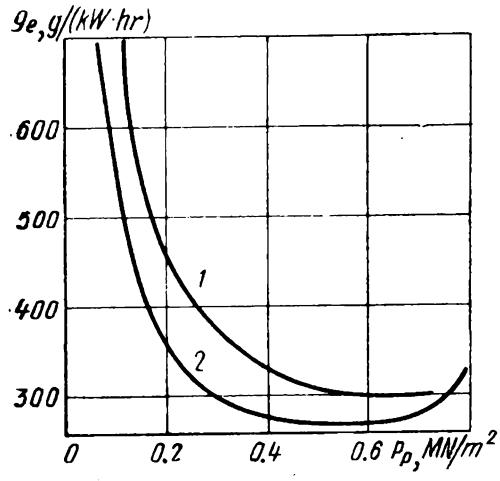

Fig. 6.9. Load characteristics of engines
1—ГАЗ-51; *2*—ГАЗ-51Ф with fuel spray ignition

6.5. Rotor-Piston Engines

The principal distinction between rotor-piston engines and internal-combustion reciprocating engines lies in the fact that the former have no crank mechanism for transforming the reciprocating motion of the pistons into the rotary motion of the crankshaft.

The attempts to design a good and economical rotor-piston engine had been undertaken long ago, but the idea was realized only some 20 years since by the engineer F. Wankel who worked with the NSU firm (FRG). The principle of action and brief description of Wankel's engine are outlined below.

The rotor-piston engine (Fig. 6.10) comprises a stationary housing *2* cooled by water with a working space *I* having the form of two mated cylinders. The housing closed by end covers accommodates rotor *3* which revolves with respect to its own axis and simultaneously runs around an immobile gear wheel *1* with the aid of an inner toothing. The rotor in the form of a triangle with arc-shaped sides is mounted on the eccentric of the output shaft. All the three apexes of the rotating rotor are in permanent contact with the walls of working space *I* and divide the working volume into three isolated chambers in which a four-stroke cycle is performed. Three flashes occur during one revolution of the rotor. The output eccentric shaft rotates three times faster than the rotor.

By their simple design, overall dimensions and mass, rotor-piston engines are much superior to internal-combustion reciprocating engines. Unbalanced inertia forces of revolving masses in these engines are equalized by simple counterweights.

Figure 6.11 shows longitudinal sections of a single-rotor (Fig. 6.11*a*) and a double-rotor (Fig. 6.11*b*) rotor-piston engines

designed by the Curtis-Wright firm which operate according to the Wankel scheme, and the front view of the engine (Fig. 6.11*c*).

The output eccentric shaft *10* carrying rotor *3*, counterweight *4* and flywheel *1* rotates in bearings *5*. Front *6* and rear *9* covers and also housing *2* are centred by pins and clamped by nuts and

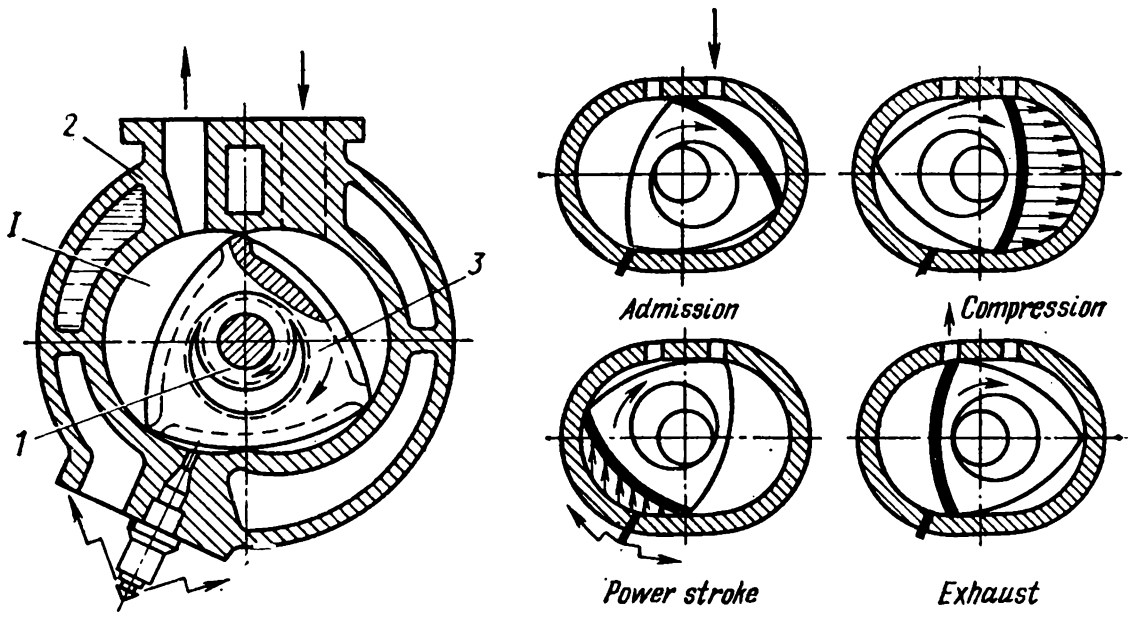

Fig. 6.10. Cross section and functional diagram of rotor-piston engine

1—working space; *1*—gear wheel; *2*—housing; *3*—rotor

studs passed through water jacket *11*. The rotor made of aluminium alloy or cast iron is cooled by oil delivered by an oil pump into duct *A* of the eccentric shaft.

The engine accommodates water *7* and oil *13* pumps, oil filter *14*, carburettor *8*, generator *12* and starter *15*.

6.6. Multi-Fuel Engines

The rapid development of diesel engines and the growing demand for heavy fuel impelled designers to seek ways and means for a combined use of surplus petrol and scarce diesel fuel. One of the solutions of this problem is to employ petrol as fuel for diesel engines, i.e., create a highly economical petrol engine.

A multi-fuel engine is a diesel engine which operates with heavy (diesel fuel, etc.) and light (petrol, etc.) petroleum fractions. If the working process of a multi-fuel high-speed diesel engine is arranged correctly, effective combustion of various motor fuels can be attained. If also conditions are provided for a appropriate mixture formation, the load on the diesel engine parts

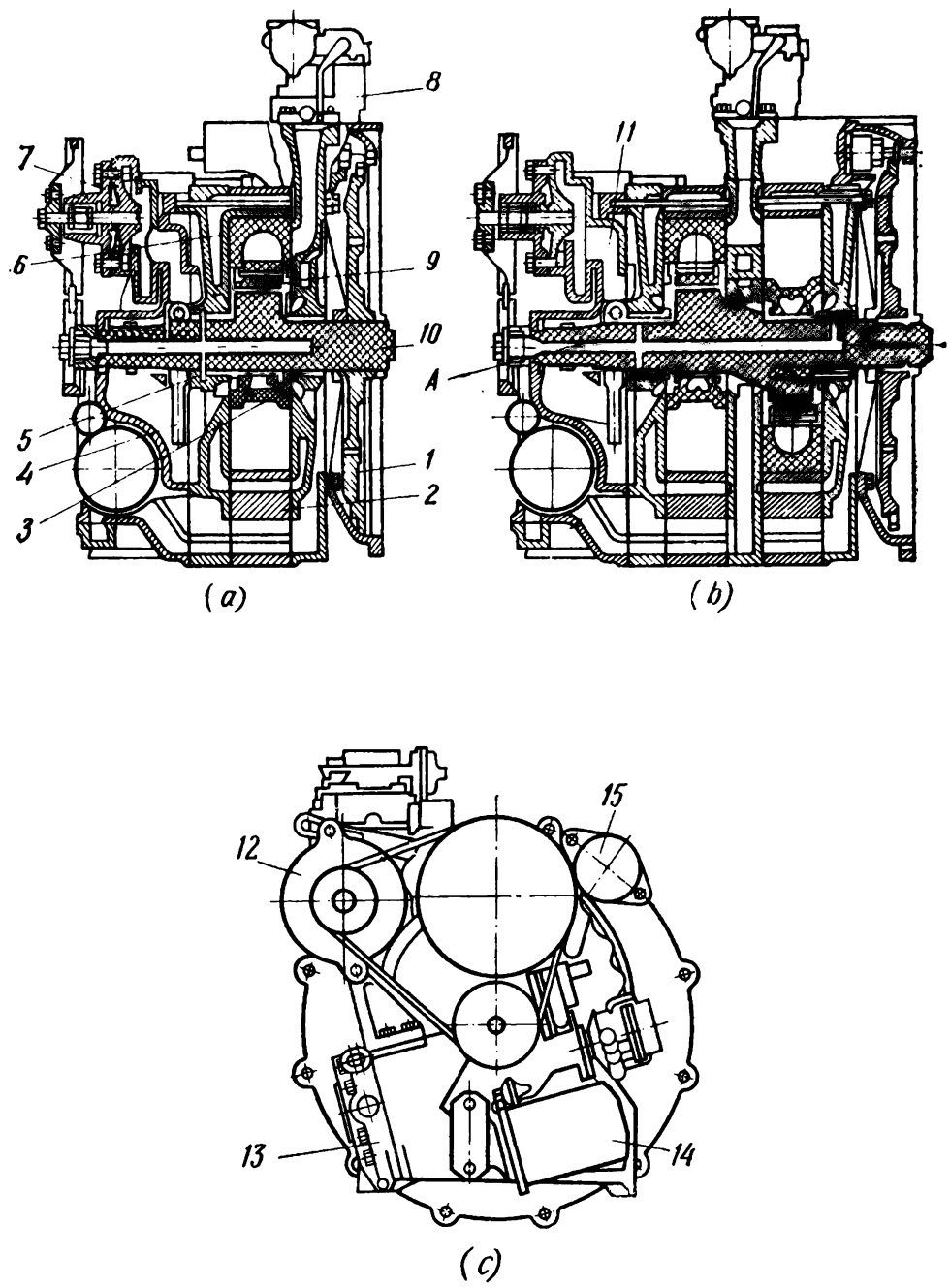

Fig. 6.11. Single- and double-rotor rotor-piston engines produced by the Curtis-Wright firm operating according to Wankel scheme

(*a*) single-rotor engine; (*b*) double-rotor engine; (*c*) front view of the engine; *A*—duct, *1*—flywheel; *2*—housing; *3*—rotor; *4*—counterweight; *5*—bearing; *6*—front cover; *7*—water pump; *8*—carburettor; *9*—rear cover; *10*—eccentric shaft; *11*—water jacket; *12*—generator; *13*—oil pump; *14*—oil filter; *15*—starter

will be increased insignificantly. The use of special combustion chambers ensures the same pressure p_z at the end of combustion for both petrol and diesel fuel.

6.7. Electric Motors

An electric motor installed on an automobile (electric car) lessens the contamination of the atmosphere by the exhaust gases, reduces the level of transport noise and simplifies the control, design and technical maintenance of a motor vehicle, because it is possible to dispense with a number of complicated units in the transmission. The characteristic of an electric motor is more favourable than the external speed characteristic of an internal-combustion engine (with a decrease in the angular velocity of the shaft the torque of an electric motor increases).

A storage battery or a battery of fuel elements can be employed on electric cars as sources of electric energy.

The fuel element is an electrochemical device in which the chemical energy of the fuel and oxidant is directly converted into electric energy.

In the existing fuel element designs, hydrogen acts as the basic fuel and pure oxygen, as the oxidant.

Thus, electric energy is obtained in a fuel element continuously during the entire time of reception of the fuel and oxidant, whereas in a galvanic cell the chemical energy is converted into electricity in the element itself. Oxidation of fuel and reduction of oxygen occur in the fuel elements electrochemically on various electrodes and the energy of reaction is obtained directly in the form of electric energy. The efficiency of a fuel element lies within 0.60-0.70 and diminishes as the load increases.

Among the principal shortcomings of present-day electric cars are large overall dimensions and unit masses of the sources of electric energy, which is the cause of a comparatively short running distance (50-70 km without recharging). The unit mass of up-to-date carburettor engines is within 2-3 kg/kW and of diesel engines within 3-7 kg/kW, whereas the unit masses of lead-acidic and iron-nickel accumulators vary within 200-250 kg/kW and of silver-zinc ones, within 40-60 kg/kW. The unit mass of fuel elements utilizing hydrogen and oxygen is within 20-30 kg/kW. It should also be noted that a high cost of silver-zinc accumulators and fuel elements which utilize hydrogen and oxygen prevents today their use on public passenger transport.

Part Two
THEORY OF MOTOR VEHICLES

Chapter VII FORCES ACTING ON A MOTOR VEHICLE

7.1. Tractive Force on the Driving Wheels

A motor vehicle in motion is acted upon by various forces — the forces which move the vehicle and the forces which offer resistance to its motion.

The principal motive force is the tractive force of the driving wheels. This force arises as a result of operation of the engine and depends on the interaction between the driving wheels and the road.

The forces of resistance include *the force of friction in the transmission, the force of resistance of the road and the force of resistance of the air.*

The tractive force P_t is determined as the ratio of the moment on the axle shaft to the radius of the driving wheels during uniform motion.

Thus, the tractive force can be estimated if the radius of the driving wheel is known. Since the wheels carry elastic pneumatic tyres, the value of the wheel radius constantly changes during motion.

Distinction is made between the following radii of a motor vehicle wheel.

Static radius of wheel r_s, i.e., distance from the road surface to the axis of a stationary wheel which takes vertical load.

Dynamic radius of wheel r_d or the distance from the road surface to the axis of a rolling wheel. The dynamic radius grows with a lesser load P_z taken by the wheel and a higher air pressure in the tyre p_t. When the speed of a motor vehicle increases under the action of the centrifugal forces the tyre expands in the radial direction and the radius r_d increases.

Rolling radius of wheel r_r is the radius of such an undeformable ring which has the same angular and linear velocity as the given elastic wheel. In a wheel, which rolls under the action of torque, the tread elements in contact with the road are compressed, and the wheel traverses a smaller path at an equal angular velocity than during free rolling. In the case of a wheel loaded with a braking torque, the tread elements coming in contact with the road are expanded. For this reason, the braked wheel traverses a larger path at the same angular velocities than a freely rolling wheel. In this way the turning moment tends to reduce the radius r_r and the braking moment, conversely, increases it.

In order to find the value of r_r, a transverse line is marked on the road with a piece of chalk or paint which, when rolled over by the wheel of a motor vehicle, leaves prints. By measuring the distance l between the extreme prints it is possible to determine the rolling radius (in m) by the formula:

$$r_r = \frac{l}{2\pi n}$$

where n is the number of wheel revolutions corresponding to the distance l.

The difference between the radii r_d and r_r is mainly caused by slip in the area where the tyre is in contact with the road. If there is no slip (driven wheel) the radii r_d and r_r are approximately equal. In case of full wheel skid the distance $l = 0$ and, therefore, the radius $r_r = 0$. When braked (nonrevolving) wheels are skidding, the number of revolutions $n = 0$ and the rolling radius grows infinitely large.

The calculations become extremely complicated if the variable values of the radius are introduced. The driving wheels rarely slip on roads with a hard pavement surface and the changes in the radius r_r are small. Therefore, the numerical values of the radii r_r, r_d and r_s are assumed equal and are denoted by the letter r.

The values of r are cited in the specifications of tyres. If they are not available, the value of r (in m) can approximately be determined from the formula:

$$r = 0.5D_r + B_t(1 - \lambda_t) \qquad (7.1)$$

where D_r = diameter of wheel rim, m
B_t = height of tyre profile in a free state, m
λ_t = radial deformation coefficient of the tyre which lies within 0.1-0.16 for standard and broad-profile tyres and within 0.2-0.3 for arched tyres and pneumatic-tyred rollers

7.2. Transmission Efficiency

Power is transmitted from the engine to the driving wheels by various units in the transmission. In this case some of the power is expended on overcoming friction between the gear teeth in the transmission gearbox and the driving axle, in the bearings and glands, and also on overcoming friction of the gears against the oil and on its splashing. For this reason, the tractive power N_t delivered to the driving wheels, when the motor vehicle runs uniformly, is less than the effective engine power N_e by the amount of the power N_f expended on friction:

$$N_t = N_e - N_f \tag{7.2}$$

One of the methods of determining the energy losses in transmission consists in the following. A balancing electric motor is installed on a motor vehicle instead of the engine and the shafts of balancing braking generators (electric brakes) are attached to the axle shafts. The electric motor rotates through transmission the shafts of the electric brakes in which resistors building up the required load are connected into the armature circuits. Having measured the angular velocities of the armatures by tachometers and the values of moments on the housings by dynamometers, it is possible to find the power N_e of the electric motor and the power N_t absorbed by the electric brakes. The difference in the power $N_e - N_t$ is the loss of power in the transmission N_f.

More often than not the losses of energy in the transmission are appraised by the value of the moment of friction M_f (in N·m) reduced to the driving wheels of a motor vehicle:

$$M_f = 1000 \frac{N_f}{\omega_w} \tag{7.3}$$

where ω_w is the angular velocity of the driving wheels in rad/s.

The values of M_f and N_f account for two kinds of losses: hydraulic losses and the losses due to friction between the teeth of gears and in universal joints. The hydraulic losses are caused by churning and splashing of oil in the transmission and driving axle cases. They hardly depend on the transmitted torque but change with variations in the angular velocity of parts, viscosity and the quantity of oil poured into the cases. Hydraulic losses are estimated by the moment M_h which would be applied to the suspended driving wheels of the motor vehicle in order to rotate idly (without load) the transmission shafts.

Figure 7.1*a* illustrates the experimental dependences of the moment M_h on the speed of motion for some types of automobiles.

In the absence of experimental data for the type 4 × 2 automobiles, use is made of the empirical formula:

$$M_h = (2 + 0.09v)\, G_a r \cdot 10^{-3} \qquad (7.4)$$

where v = speed of automobile, m/s
G_a = full weight (force of gravity) of automobile, N
r = radius of driving wheels, m

The losses of energy on friction may be assumed proportional to the moment transmitted by the transmission and independent

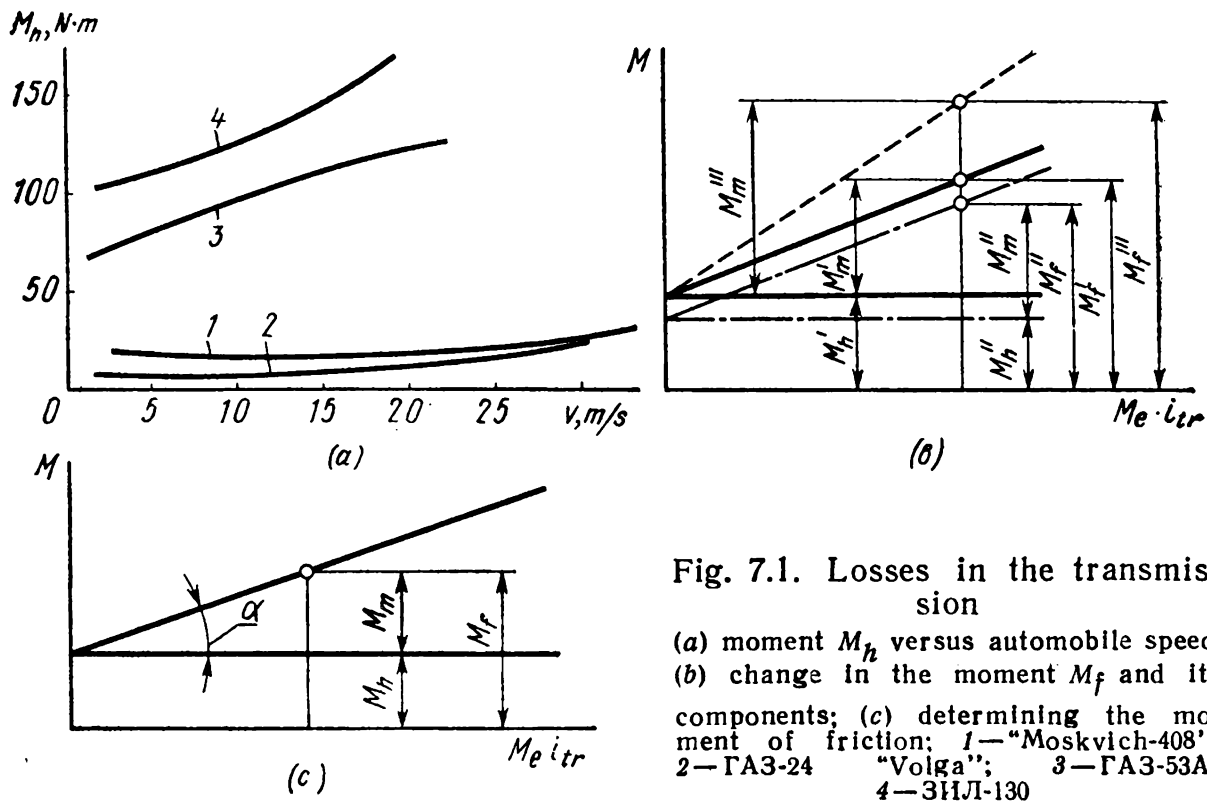

Fig. 7.1. Losses in the transmission

(*a*) moment M_h versus automobile speed; (*b*) change in the moment M_f and its components; (*c*) determining the moment of friction; *1*—"Moskvich-408"; *2*—ГАЗ-24 "Volga"; *3*—ГАЗ-53А; *4*—ЗИЛ-130

of the rotation speed of parts. These losses are estimated by the moment M_m.

The general nature of the dependences of the total moment of friction M_f in the transmission and its components M_h and M_m on the transmitted torque is shown in Fig. 7.1*b*. The horizontal straight lines describe the value of M_h and the inclined lines, the value of $M_f = M_h + M_m$. Let the resistances M'_m, M'_h M'_f represented by the solid lines correspond to some period of time. If the reduction in oil viscosity or speed of rotation diminishes the hydraulic losses to M''_h, the total moment will become equal to M''_f (dash-and-dot lines). If the losses in the engagement of gears increase and the moment M_m rises to M'''_m (dash line), the total moment will prove equal to M'''_f.

In conformity with Fig. 7.1*c* it may be written that

$$M_f = M_h + M_m = M_h + M_e i_{tr} \tan \alpha_t = M_h + M_e i_{tr} (1 - \chi) \quad (7.5)$$

where α_t = inclination angle of the straight line to the X-axis

$\chi = (1 - \tan \alpha_t)$ = coefficient of the effect of load whose value can be determined from the empirical formula:

$$\chi = 0.98^k \times 0.97^l \times 0.99^m \quad (7.6)$$

k, l, m are the numbers of pairs of spur and bevel gears and number of universal joints transmitting the torque.

If the moment M_f is known, it is possible to find the *force of transmission resistance* (in N)

$$P_{tr} = \frac{M_f}{r}$$

Various methods of estimating losses are employed depending on the operating conditions of the transmission. Thus, if energy is transmitted by the transmission from the engine to the driving wheels, the *direct transmission efficiency* η_{tr} is used which is the ratio of the tractive power on the driving wheels N_t to the effective power or their corresponding moments:

$$\eta_{tr} = \frac{N_t}{N_e} = \frac{N_e - N_f}{N_e} = \frac{M_e i_{tr} - M_f}{M_e i_{tr}} \quad (7.7)$$

where i_{tr} is the gear ratio of the transmission.

When a motor vehicle is braked by the engine the transmission transmits energy from the driving wheels to the engine and the energy losses are estimated by the *reverse transmission efficiency*:

$$\eta_{rev} = \frac{N_{f.e}}{N_{f.e} + N_f} = \frac{M_{f.e} i_{tr}}{M_{f.e} i_{tr} + M_f} \quad (7.8)$$

where $N_{f.e}$ is the power expended on friction in the engine (braking power), kW.

Since in recalculations the efficiency η_{tr} is used more often than the efficiency η_{rev}, the word "direct" is usually omitted in the definition of the transmission efficiency.

When a motor vehicle moves under inertia and the engine is disengaged from the transmission ($i_{tr} = 0$), expression (7.7) has no sense and the losses of energy in the transmission are appraised from their absolute value, i.e., from the value of the moment M_f or power N_f.

It follows from (7.5) and (7.7) that

$$\eta_{tr} = \chi - \frac{M_h}{M_e i_{tr}} \quad (7.9)$$

Since

$$M_e i_{tr} = M_t + M_f = \frac{M_t + M_h}{\chi}$$

expression (7.9) may be written as follows:

$$\eta_{tr} = \chi\left(1 - \frac{M_h}{M_t + M_h}\right)$$

Example. Determine the value of η_{tr} for a car in direct drive ($i_{tr} = 5.0$, $k = 0$) and first speed ($i_{tr} = 15$, $k = 2$) in the transmission with the following parameters: $M_e = 100$ N·m; $M_h = 20$ N·m; $l = 1$ (single final drive); $m = 3$.

In direct drive:

$$\chi = 0.98^k \times 0.97^l \times 0.99^m = 0.97 \times 0.99^3 = 0.94$$

$$\eta_{tr} = \chi - \frac{M_h}{M_e i_{tr}} = 0.94 - \frac{20}{100 \times 5} = 0.90$$

In first speed:

$$\chi = 0.98^2 \times 0.97 \times 0.99^3 = 0.90$$

$$\eta_{tr} = 0.90 - \frac{20}{100 \times 15} \approx 0.89$$

When an engine operates under average load, for example, when a motor vehicle runs over a horizontal smooth road, the value of η_{tr} may be much less than χ and noticeably changes with the change in the speed of the motor vehicle. If the engine operates under full load the value of M_h is several times less than the product $M_e i_{tr}$ and their ratio may therefore be neglected assuming that $\eta_{tr} \approx \chi$. It is surmised in this case that the transmission efficiency changes only upon the change in the number of gear couples in the transmission which transmit the torque. This change is disregarded in approximate calculations assuming the coefficient η_{tr} to be a constant magnitude.

The tentative values of direct (for an engine operating under full load) and reverse efficiency of the transmission are illustrated in Table 7.1.

TABLE 7.1

Transmission Efficiency

Motor vehicles	η_{tr}	η_{rev}
Sport-racing cars	0.90-0.95	0.80-0.85
Motor cars	0.90-0.92	0.80-0.82
Trucks and buses	0.82-0.85	0.75-0.78
Cross-country vehicles	0.80-0.85	0.73-0.76

7.3. Traction Characteristic of a Motor Vehicle

The traction characteristic of a motor vehicle is a graphically represented dependence of the tractive force on the motor vehicle speed. The traction characteristic is plotted from the results of stand or road tests or from calculated data. In order to carry out stand tests, car *1* (Fig. 7.2) is placed with its driving wheels on running drums *4* of the stand and fastened by means of a wire rope through dynamometer *2* to stationary post *3*. At full throttle a hydraulic or electric brake is used to build up such resistance to the rotation of the running drums when their angular velocity

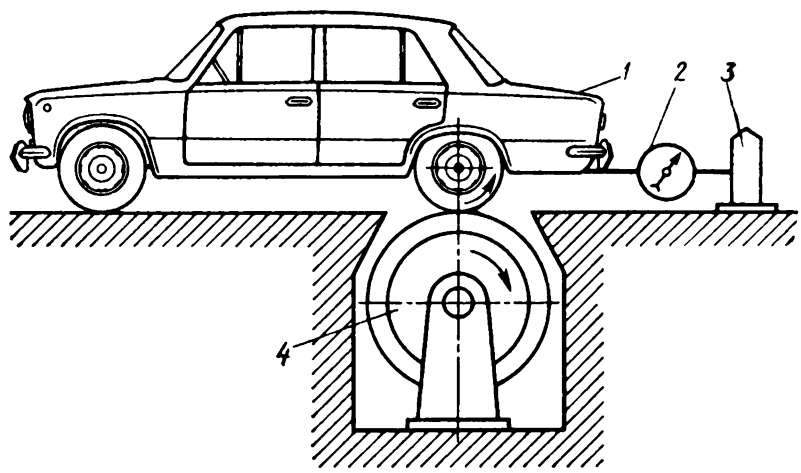

Fig. 7.2. Stand with running drums
1—car; *2*—dynamometer; *3*—post; *4*—running drum

remains constant. Having measured with a tachometer the angular velocity ω_w of the driving wheels and knowing their radius it is possible to find the speed (in m/s) at which the car would move on a road:

$$v = \omega_w r \tag{7.10}$$

The angular velocities of the driving wheels ω_w and crankshaft ω_e are interrelated by the equality:

$$\omega_e = \omega_w i_{tr} = \omega_w i_g i_{aux} i_{main}$$

where i_g, i_{aux}, i_{main} are the gear ratios of the gearbox, auxiliary transmission and main gearbox, respectively.

Hence,

$$v = \frac{\omega_e r}{i_{tr}} \tag{7.11}$$

If the energy loss is neglected, when the driving wheels roll on the stand drums, it may be assumed that the driving force is

equal to the force loading the dynamometer which shows the magnitude of the force P_t. Sometimes the dynamometer is connected to the shaft of the running drums and measures the torque directly on this shaft.

During road tests, dynamometers are employed to determine the torque on the universal-joint shaft or on the axle shaft of the motor vehicle. With the same aim in view tensometers are bonded to the shaft or axle shaft, which record even short-time changes in the moment on the tape of an oscillograph.

If the experimental information is not available, the force P_t is determined by calculation using for the purpose the speed characteristic of the engine.

As the moment M_e is transmitted by the units of transmission, its value changes in proportion to the gear ratios of the units. The moment (in N·m) delivered to the axle shaft during uniform motor vehicle motion

$$M_t = M_e i_{tr} - M_f = M_e i_g i_{aux} i_{main} - M_f \tag{7.12}$$

With regard to formula (7.7) the moment M_t can also be found as follows:

$$M_t = \eta_{tr} M_e i_{tr} \tag{7.13}$$

and the tractive force (in N) from the formula:

$$P_t = \frac{M_t}{r} = \frac{M_e i_{tr} - M_f}{r} = \eta_{tr} \frac{M_e i_{tr}}{r} \tag{7.14}$$

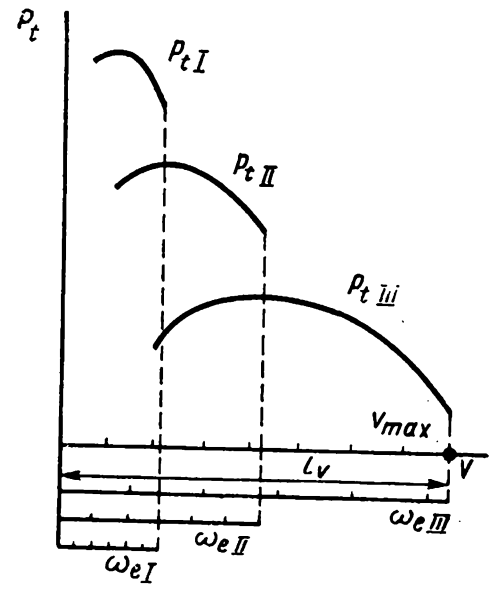

Fig. 7.3. Traction characteristic of a motor vehicle

Having determined M_e, η_{tr} (or M_f) and v for several values of ω_e, it is possible with the aid of formula (7.14) to find the dependence of the tractive force on the motor vehicle speed within the entire range of change in the angular velocity ω_e and the moment M_e, and plot a traction characteristic. The number of curves on this graph corresponds to the number of steps in the gearbox (Fig. 7.3).

The scales of angular velocities ω_e with different gears are shown at the bottom of the graph. First, the length l_v on the speed scale is determined from the dimensions of the graph and the maximum speed v_{max}. The same length l_v should also be on the scale of angular velocity at the highest gear. This scale is drawn below the scale of the speed v. At a constant angular velocity of the engine the speed of the motor vehicle at various gears is

inversely proportional to the gear ratios of the gearbox:

$$v_I : v_{II} = i_{II} : i_I,$$
$$v_{II} : v_{III} = i_{III} : i_{II} \ldots \text{ etc.}$$

For this reason, the division on the scale of angular velocity is, for example, for the first gear i_I times less than the division on the scale for direct drive.

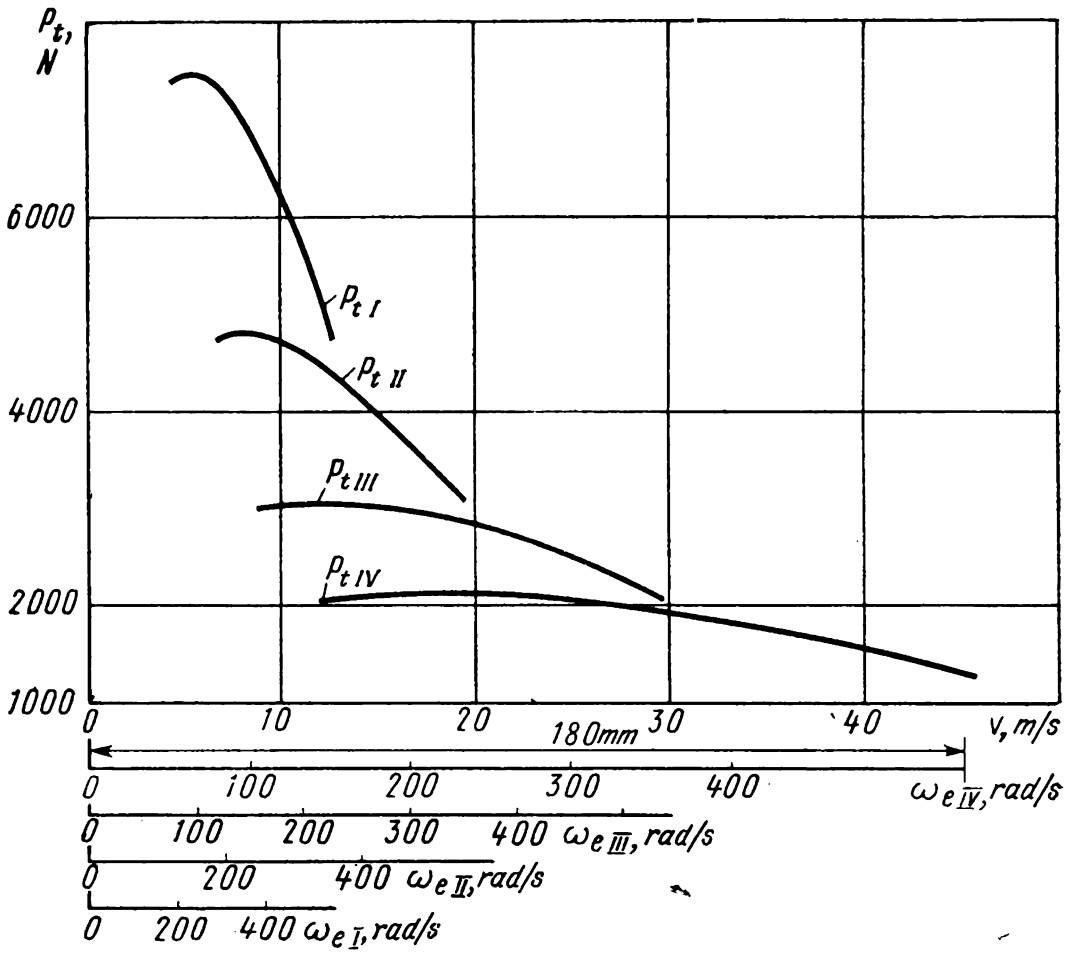

Fig. 7.4. Traction characteristic of a motor vehicle accepted for calculation

Example. Plot the traction characteristic for a car from the following data: $i_I = 3.51$; $i_{II} = 2.26$; $i_{III} = 1.45$; $i_{IV} = 1.00$; $i_{main} = 4.1$; $r = 0.33$ m; $\eta_{tr} = 0.9$. The values of torque M_e are tabulated in the example of plotting the speed characteristic of a four-stroke carburettor engine.

When the car runs in first gear at $\omega_e = 136$ rad/s and $M_e = 184$ N·m, we get

$$v = \frac{\omega_e r}{i_g i_{main}} = \frac{136 \times 0.33}{3.51 \times 4.1} = 3.12 \text{ m/s}$$

$$P_t = \eta_{tr} \frac{M_e i_g i_{main}}{r} = \frac{0.9 \times 184 \times 3.51 \times 4.1}{0.33} = 7230 \text{ N}$$

The results of calculating P_t and v for other values of ω_e, M_e and i_g are shown in the table and the traction characteristic of the calculated car is represented in Fig. 7.4.

Parameters	Angular velocity of crankshaft, rad/s					
	136	220	304	388	472	556
v_{I}, m/s	3.12	5.05	7.00	8.90	10.85	12.80
v_{II}, m/s	4.85	7.85	10.80	13.80	16.80	19.80
v_{III}, m/s	7.55	12.20	16.90	21.50	26.20	30.90
v_{IV}, m/s	11.00	17.70	24.50	31.20	38.00	44.80
$P_{t\mathrm{I}}$, N	7230	7450	7390	6820	5960	4750
$P_{t\mathrm{II}}$, N	4660	4810	4760	4400	3850	3030
$P_{t\mathrm{III}}$, N	2980	3080	3050	2820	2460	1960
$P_{t\mathrm{IV}}$, N	2060	2120	2100	1940	1700	1350

If a scale of 4 mm = 1 m/s is selected for the speed scale, then at 45 m/s the length of the scale will be $l_v = 180$ mm. The same length describes the scale $\omega_{e\mathrm{IV}}$ whose extreme division corresponds to the angular velocity of the crankshaft:

$$\omega_{e\mathrm{IV}} = \frac{v i_g i_{main}}{r} = \frac{45 \times 1 \times 4.1}{0.33} = 558 \text{ rad/s}$$

The lengths of the scales of the crankshaft angular velocities are respectively equal to $l_{\mathrm{III}} = l_v : i_{\mathrm{III}} = 180 : 1.45 = 124$ mm; $l_{\mathrm{II}} = 180 : 2.26 \approx 80$ mm; $l_{\mathrm{I}} = 180 : 3.51 \approx 51$ mm.

The graph of the traction characteristic corresponds to the case of uniform motion of a motor vehicle, which occurs rather rarely. More often than not a motor vehicle moves with acceleration or deceleration. In such cases the tractive force must not be estimated from formula (7.14), since its magnitude is acted upon by the forces of inertia of the engine rotating parts. Nonuniform rotation of the crankshaft causes an inertia moment (mainly of the flywheel) which is in opposition to the direction of angular acceleration or deceleration of the flywheel. When the motor vehicle is accelerated, this moment is equal to the product $I_f \varepsilon_f$, where I_f is the moment of inertia of the flywheel in N·m·s^2 and ε_f is its angular acceleration in rad/s^2.

If small moments of inertia of the gears and transmission shafts are disregarded, the moment M_{ax} (in N·m) on the axle shafts during acceleration can be found from the formula:

$$M_{ax} = \eta_{tr} (M_e - I_f \varepsilon_f) i_{tr} \qquad (7.15)$$

The total moment of the inertia force from acceleration of the driving wheels is equal to the product $I_2 \varepsilon_2$, where I_2 is the total

moment of inertia of the driving wheels in N·m·s² and ε_2 is their angular acceleration, rad/s² *.

Hence, the moment (in N·m) on the driving wheels in acceleration is

$$M_{ax} = \eta_{tr} (M_e - I_f \varepsilon_f) i_{tr} - I_2 \varepsilon_2 \quad (7.16)$$

On the basis of kinematic relationships it may be written that

$$\varepsilon_2 = \frac{j}{r}, \qquad \varepsilon_f = \varepsilon_2 i_{tr} = \frac{j}{r} i_{tr} \quad (7.17)$$

Inserting the values of ε_2 and ε_f into (7.16), we obtain

$$M_{ax} = \eta_{tr} M_e i_{tr} - \frac{\eta_{tr} I_f i_{tr}^2 + I_2}{r} j \quad (7.18)$$

where j is the acceleration of the wheel centre, equal in linear motion to the acceleration of the gravity centre of the motor vehicle, in m/s².

In this case, the peripheral force P_{ax} (in N) acting on the driving wheels of an automobile in acceleration can be found from the formulas:

$$P_{ax} = \frac{M_{ax}}{r} = \eta_{tr} \frac{M_e i_{tr}}{r} - \frac{\eta_{tr} I_f i_{tr}^2 + I_2}{r^2} j \quad (7.19)$$

$$P_{ax} = P_t - \frac{\eta_{tr} I_f i_{tr}^2 + I_2}{r^2} j \quad (7.20)$$

It follows from the latter equation that when a motor vehicle is accelerated ($j > 0$), the peripheral force P_{ax} on the driving wheels is less than the tractive force P_t due to the energy being expended to accelerate the engine flywheel and the driving wheels of the automobile. Conversely, during deceleration ($j < 0$) the peripheral force is higher than the tractive force and the energy accumulated during acceleration can be used to continue the motion.

7.4. Road Resistance

The interaction between a motor vehicle and road involves an energy expenditure which can be divided into three groups: energy spent on traction, on deformation of tyres and road, and on vibration of the automobile parts.

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* Here and further the index 2 denotes the values which refer to the rear driving wheels and the index 1 — to the front driven wheels.

Resistance Due to Grade

Motor roads usually have many hills. The steepness of an ascent is described by the angle α_{rd} in degrees or by the road gradient i, which represents the ratio of H to B (Fig. 7.5).

Let us decompose the gravity force G of a motor vehicle climbing uphill at an angle of α_{rd} into two components: the force $G \sin \alpha_{rd}$ parallel to the road and the force $G \cos \alpha_{rd}$ perpendicular to it.

The force $G \sin \alpha_{rd}$ is known as the *force of resistance due to grade* and is denoted by the letter P_g. The angles of gradient are

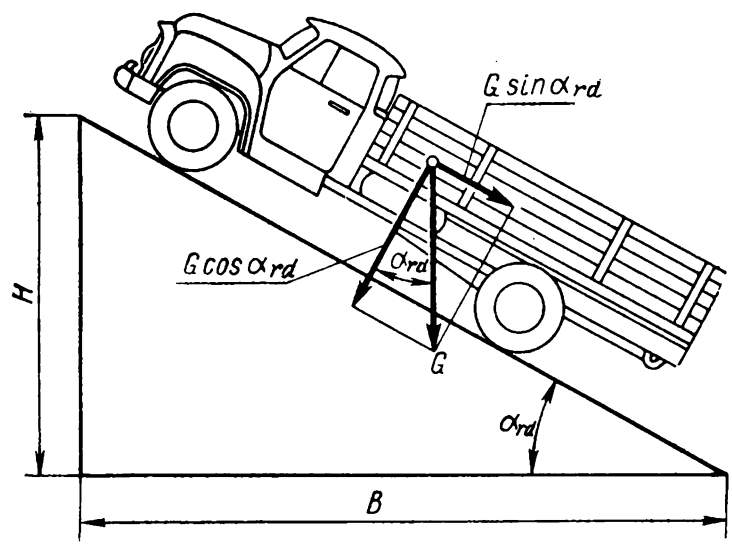


Fig. 7.5. Force of resistance due to grade

small on motor roads with a hard pavement and do not exceed 4-5°. It may be assumed for such angles that one hundredth fraction of the gradient corresponds to 35′ of the angle α_{rd}. In this case, the graident $i = \tan \alpha \approx \sin \alpha$. Then, the force (in N) is

$$P_g = G \sin \alpha_{rd} \approx i\,(G) \qquad (7.21)$$

The power expended on an ascent with gradient i (in kW) is

$$N_g = \frac{P_g v}{1000} \approx \frac{iGv}{1000} \qquad (7.22)$$

Example. Find the force and power of resistance due to grade of a car travelling at 10 m/s on a gradient angle of 3° 30′. The weight of the car G = 17,900 N.

The road gradient at $\alpha_{rd} = 3° 30'$

$$i = \tan 3°30' = 0.06$$

The force of resistance due to grade at any speed

$$P_g = iG = 0.06 \times 17{,}900 \approx 1070 \text{ N} = 1.07 \text{ kN}$$

The power of resistance due to grade at a speed of 10 m/s

$$N_g = \frac{P_g v}{1000} = \frac{1070 \times 10}{1000} = 10.7 \text{ kW}$$

In motion during descent the force P_g is directed along the motion and is the motive force. For this reason, the angle α_{rd} and the road gradient i are positive if a motor vehicle moves uphill and negative if it moves downhill.

As a rule, modern motor roads have no clear-cut portions with a constant gradient: their longitudinal profile is shaped smoothly. The gradient and the force P_g constantly change as the motor vehicle moves.

Rolling Resistance

The rolling resistance depends on the deformation of tyres and road, the friction of tyres on the road surface and friction in the wheel bearings. The deformation of tyres and road are paramount.

A tyre comes in contact with the road at an infinitely large number of points. At each point the tyre is acted upon by an infinitely small force — the elementary reaction of the road. *The resultant of the elementary forces acting from the road on the wheel in the area of contact is called the reaction of the road.* The road spatial reaction can be represented by three components: a normal component Z perpendicular to the road, a tangential component X acting in the plane of the road and in the plane of the wheel, and a crosswise component Y lying in the plane of the road and perpendicular to the plane of the wheel. These road reaction components Z, X and Y are applied to the tyre in the area of contact. The reactions X and Y can appear only if the wheel is acted upon by the reaction Z.

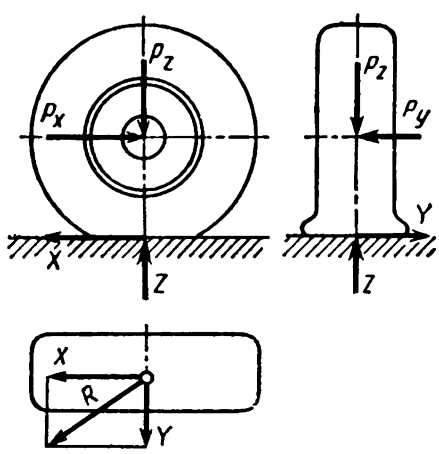


Fig. 7.6. Road reactions acting on the wheel

If the wheel is affected by the longitudinal P_x and transverse P_y forces (Fig. 7.6), a horizontal road reaction appears in the area of contact

$$R = \sqrt{P_x^2 + P_y^2} = \sqrt{X^2 + Y^2}$$

The force of road tyre grip R_{gr} is the maximum value of the horizontal reaction R_{max} proportional to the vertical load on the wheel:

$$P_{gr} = R_{max} = \varphi Z$$

where φ is the grip coefficient.

The grip coefficient φ is numerically equal to the ratio of the force causing uniform wheel slip to the normal road reaction. Distinction is made between *longitudinal* φ_x and *transverse* φ_y *grip* coefficients depending on the direction of the wheel slip. The values of the coefficients φ_x and φ_y depend on the type and state of the road, the pattern and the degree of wear of the tread, the air pressure in the tyre and also on the speed of motion and the normal load taken by the wheel.

The grip coefficient is often identified with the coefficient of sliding friction. This is not quite exact, however, because not only

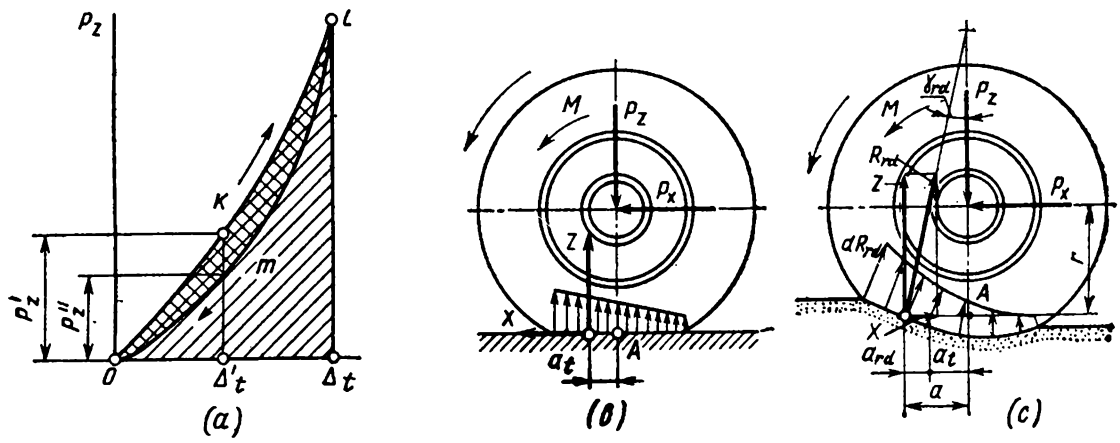


Fig. 7.7. Rolling resistance
(*a*) hysteresis losses in the tyre; (*b*) rolling of a wheel on hard road; (*c*) rolling of a wheel on deformable road

friction but also mechanical cohesion of the surfaces occurs as the tyre runs on the road. The following condition should be observed for a wheel to move without longitudinal and transverse slip:

$$P_{gr} = \varphi Z \geqslant \sqrt{X^2 + Y^2} \qquad (7.23)$$

When an elastic wheel rolls on a horizontal hard road, the elementary normal road reactions and their resultant are arranged vertically. During rolling, friction arises between parts of the tyre, the liberated heat is dissipated and the work expended on tyre deformation is not regained completely as the tyre returns to its original form. As the vertical load P_z increases, the tyre deformation Δ_t is measured to obtain the dependence shown in Fig. 7.7*a* as curve *Okl*. If the vertical load decreases, lesser load values (curve *lmO*) correspond to the same tyre deformations. The area of the loop *OklmO* represents, to a definite scale, the work pertaining to the irreversible losses in the tyre (hysteresis). As the wheel rolls, the deformations increase in the front part of the tyre and diminish in the rear portion. Therefore, the portions of the tyre at equal distance to the front and rear from the vertical diameter of the wheel and displaying identical defor-

mations are acted upon by different forces. Thus, with one and the same deformation Δ_t' the portion of the tyre in the front is loaded with the force P_z' and the portion at the rear, with the force P_z''. Hence, the elementary normal reactions in the front area of contact are larger than at the rear (Fig. 7.7*b*), and the line of action of the resultant Z is displaced with respect to the vertical diameter forward by a distance of a_t.

The displacement a_t is mainly caused by internal tyre friction. It grows with a larger number of cord plies, the greater thickness of the tread and the width of the wheel rim, and diminishes with a higher tyre air pressure.

The displacement of the normal reaction generates the moment Za_t. In order to equalize this moment, an equal but opposed moment should be applied to the wheel, or a pushing force P_x forming force couple together with the tangential reaction of the road to the wheel axis. The force

$$P_x = \frac{a_t}{r} Z$$

When a deformable wheel rolls on a soft road (Fig. 7.7*c*) energy is spent on overcoming the tyre internal friction, deformation of the road and friction between the tyre and the soil. The tyre plunges into the soil, squeezes it sideways and compacts separate particles forming a track. The elementary reactions dR_{rd} at each point of the tread are inclined perpendicular to the contact area. Therefore, the total reaction R_{rd} is inclined to the vertical at a certain angle γ_{rd}. Principally, the work of compacting the road is performed when the tyre rolls onto deformable soil, this increases the reactions dR_{rd} in the front portion of contact and diminishes them in the rear portion. As a result, the road reaction R_{rd} is displaced forward with respect to the vertical diameter by distance a. This displacement can be represented as the displacement a_t caused by the hysteresis losses in the tyre, and the displacement a_{rd} due to the energy being spent to form a track.

If a wheel rolls under the action of the pushing force P_x, the following can be obtained from the equation of moments of all forces with respect to point A:

$$P_x = \frac{a}{r} Z$$

The ratio $\frac{a}{r}$ *is called the coefficient of rolling resistance and is denoted by the letter* f. In conformity with the previous equality

$$f = \frac{a}{r} = \frac{P_x}{Z}$$

The coefficient of rolling resistance is numerically equal to the ratio of the force causing uniform rolling of the wheel to the normal reaction of the road.

The force of rolling resistance P_r (in N) is the product fZ:

$$P_r = fZ \tag{7.24}$$

The moment of rolling resistance M_r (in N·m) is the product $P_r r$:

$$M_r = P_r r = fZr \tag{7.25}$$

Thus, when a driven wheel rolls uniformly, the tangential reaction of the road acting on it

$$X_1 = fZ_1 = P_{r1}$$

If a driving wheel rolls uniformly under the action of torque, the reaction of the road acting on it

$$X_2 = \frac{M_t}{r} - P_{r2} = P_t - P_{r2} \tag{7.26}$$

In the case of nonuniform rolling of the rear driving wheels, they should be acted upon by the moment equal not to the moment M_t but to the M_{ax} whose magnitude also depends on the moments of inertia of the rotating parts of the motor vehicle and the rate of its acceleration [equation (7.18)]. For this reason, the tangential reaction (in N) acting on the driving wheels of the motor vehicle in this case

$$X_2 = \frac{M_{ax}}{r} - P_{r2} = P_t - \frac{\eta_{tr} I_f i_{tr}^2 + I_2}{r^2} j - P_{r2} \tag{7.27}$$

The front driven wheels do not take the torque. Hence, if $i_{tr} = 0$ is assumed in expression (7.27) the formula will be obtained for determining the tangential reaction applied to the front wheels of the motor vehicle:

$$X_1 = -\left(\frac{I_1}{r^2} j + P_{r1}\right) \tag{7.28}$$

where I_1 is the total moment of inertia of the front wheels in $N \cdot m \cdot s^2$.

The resistance offered to tyre rolling on the road is the consequence of the expenditure of energy on the hysteresis (internal) losses in the tyre and also to form the track and on surface friction (external losses). Since it is extremely difficult to take all

these factors into account the rolling resistance is usually evaluated by the total expenditure of energy, assuming arbitrarily the force of rolling resistance as external, with respect to the motor vehicle.

In a laboratory, the coefficient f is determined from test stands using running drums as shown in Fig. 7.2. However, they are employed to test not the whole car but a separate wheel rotated by an electric motor. During road tests, a truck tractor tows a freely running automobile or a special bogie linked with the tractor by means of a wire rope and a dynamometer. To preclude air resistance, the test is conducted at a low speed (about 3 m/s).

The force (in N) loading the dynamometer

$$P_x = f_1 Z_1 + f_2 Z_2$$

where Z_1 and Z_2 = normal reactions on the front and rear axles of the towed automobile, N

f_1 and f_2 = rolling resistance coefficients of the front and rear wheels

If it is assumed that $f_1 = f_2 = f$, the force of rolling resistance (in N)

$$P_r = f(Z_1 + Z_2) = fG \qquad (7.29)$$

The power (in kW) required to overcome the rolling resistance when a motor vehicle moves at speed v is

$$N_r = \frac{P_r v}{1000} = \frac{fGv}{1000} \qquad (7.30)$$

The so-called running-out method is also frequently used to estimate the coefficient f. As a motor vehicle moves, the driver disengages the drive and the vehicle travels under inertia. At a low speed the kinetic energy of the motor vehicle is primarily expended to overcome the rolling resistance. By measuring the path or the time of motion of a motor vehicle within a certain range of speeds, it is possible to find the coefficient f (see Sec. 9.5).

At a low speed (up to 14-16 m/s) the coefficient of rolling resistance may be regarded as constant. Motion at a higher speed noticeably increases the coefficient f because the tyre has no time to straighten out completely at the contact area and not all the energy expended on deforming the tyre is recovered. Besides, a higher deformation rate increases the internal friction in the tyre casing, which also increases the coefficient f. Depending on the

speed, its magnitude is determined from the empirical formula:

$$f = f_0\left(1 + \frac{v^2}{1500}\right) \tag{7.31}$$

where f_0 is the rolling resistance coefficient at a low speed (Table 7.3).

TABLE 7.3

Coefficient of Rolling Resistance

Type of road	f_0 at $v = 14$ m/s	f (mean value)
Asphalt-concrete and cement-concrete roads:		
in excellent state	0.014	0.014-0.018
in satisfactory state	0.018	0.018-0.020
Cobblestone pavement	0.025	0.023-0.030
Gravel road	0.02	0.020-0.025
Dirt road:		
dry compact	—	0.025-0.035
after rain	—	0.05 -0.15
Sand	—	0.10 -0.30
Compacted snow	—	0.07 -0.10

Example. Find the force and power of the rolling resistance of a car ($G = 17.9$ kN) moving at various speeds, if $f_0 = 0.014$.

The force of rolling resistance:

at $v \leqslant 14$ m/s

$$P_r = f_0 G = 0.014 \times 17{,}900 \approx 250 \text{ N}$$

at $v = 20$ m/s

$$P_r = f_0\left(1 + \frac{v^2}{1500}\right) G = 0.014\left(1 + \frac{400}{1500}\right) 17{,}900 = 318 \text{ N}$$

The power of rolling resistance:

at $v = 10$ m/s

$$N_r = \frac{P_r v}{1000} = \frac{250 \times 10}{1000} = 2.5 \text{ kW}$$

at $v = 20$ m/s

$$N_r = \frac{318 \times 20}{1000} = 6.36 \text{ kW}$$

If a motor vehicle runs on hard road surface the coefficient f increases with a drop of air pressure in the tyre. When torque is transmitted, the coefficient f slightly grows, since in this case the tyre is deformed not only in a vertical direction but also on the periphery. If the transmitted torque is high, the tread elements slip on the road and additional power is expended on friction in the area of contact.

A wheel rolling on soft soil compacts the soil particles and

displaces them sideways and in the direction of motion. In this case, the coefficient of rolling resistance is affected by the depth of track, type and state of soil, wheel diameter and the vertical load it takes. The coefficient C_s showing the resistance of the soil to compression at a depth of 1 cm from the surface depends on the type and state of the soil. The values of C_s are illustrated in Table 7.4.

TABLE 7.4

Values of Coefficient C_s

Soil	Relative humidity of soil		
	0.5 (dry)	0.5-1.0 (flaky)	1.0 (loose)
Sand	0.15-0.5	—	—
Sandy loam	1.0-1.5	0.2-0.6	0.05-0.1
Clayey	1.0-2.0	0.1-0.5	0.05-0.1
Gravel mixture	—	0.5-1.0	—

If the value of C_s is known, the coefficient f can be determined from the empirical formula: $f = \frac{0.12}{C_s} + 0.05$.

A drop of air pressure in the tyre reduces the depth of the track but increases the tyre internal losses. Therefore, an internal air pressure value can be found for each tyre, at which the coefficient f will be minimum for any given soil.

Road Roughness Resistance

Any paved surface is never absolutely smooth. Even new cement and asphalt-concrete road surfaces have irregularities about 1 cm high. The roughness produces additional resistance to the motion of motor vehicles making their axles, wheels and bodies oscillate. During these oscillations, friction dissipates energy in the tyre and parts of the suspension; dissipation of energy may be rather high.

The shape, dimensions and arrangement of irregularities are random and cannot be calculated analytically. The smoothness of a paved surface is therefore found experimentally. For this purpose, the shockmeter designed by A. K. Birul is frequently installed on a motor vehicle. As the latter runs, the vertical motions of the rear axle caused in traversing irregularities are transmitted to a toothed rack linked with a counter through a pawl coupling. The coupling transmits the torque only in one direction and the counter, therefore, totals all the spring compressions during the run. The accepted standards are the values shown by the shock-

meter installed on a ГАЗ-20 "Pobeda" car moving with two passengers on the rear seat at a speed of 14 m/s. The smoothness of various types of pavement is illustrated in Table 7.5.

TABLE 7.5

Smoothness of Various Types of Pavement

Type of pavement	Readings of shockmeter in cm/km with state of pavement		
	excellent	good	bad
Asphalt- and cement-concrete	50-75	150	300
Broken stone and gravel	200	350-400	800-900
Cobblestone	300	500	1000

The additional resistance caused by the roughness of the road is taken into account by increasing conditionally the rolling resistance coefficient. For this purpose, the following empirical formula is used:

$$f = 0.01 + 1.3 \times 10^{-7} \lambda s_{sh} v^2 \qquad (7.32)$$

where λ = coefficient which depends on the design of the running gear of a motor vehicle (4 for cars and 5.5 for trucks)

s_{sh} = readings of the shockmeter, cm/km

Mean values of the coefficient f are used in the calculations because many factors affect this coefficient (see Table 7.3).

Road Resistance

When a motor vehicle moves uphill or downhill the component of the vehicle weight perpendicular to the road is equal to $G \cos \alpha_{rd}$. For this reason, the rolling resistance P_r during motion on such road sections is equal to $fG \cos \alpha_{rd}$, i.e., it is slightly less than in motion on a horizontal road. But, $\cos \alpha_{rd} \approx 1$ for small angles and this makes it possible to estimate the force P_r from formula (7.29) and also for non-horizontal sections of the road.

The magnitudes of the rolling resistance coefficient f and road gradient i describe in combination the type and state of the road pavement and also its longitudinal profile. The notion of the *force of road resistance* P_{rd} is introduced equal to the sum of the forces P_r and P_g:

$$P_{rd} = P_r + P_g = (f \cos \alpha_{rd} + \sin \alpha_{rd}) G \approx (f + i) G \qquad (7.33)$$

The expression in the brackets is known as the *coefficient of road resistance* and denoted by the letter ψ. Therefore, the force of road resistance (in N)

$$P_{rd} = \psi G \tag{7.34}$$

and the *power of road resistance* (in kW)

$$N_{rd} = \frac{P_{rd} v}{1000} = \frac{\psi G v}{1000} \tag{7.35}$$

Example. Find the force and power of road resistance for a car travelling at a speed of 10 m/s on a road with a good asphalt pavement if the angle of road gradient is 3° 30′. The rolling resistance coefficient $f = 0.02$.

The road resistance coefficient

$$\psi = f \cos \alpha_{rd} + \sin \alpha_{rd} = 0.02 \times 0.998 + 0.061 = 0.081$$

The road resistance force

$$P_{rd} = \psi G = 0.081 \times 1790 = 1450 \text{ N}$$

The road resistance power

$$N_{rd} = \frac{P_{rd} v}{1000} = \frac{1450 \times 10}{1000} = 14.5 \text{ kW}$$

The resistance force due to grade always acts simultaneously with the rolling resistance force. Therefore, the coefficient ψ which takes into account both the forces is highly important in practical calculations.

7.5. Air Resistance

The motion of an automobile is associated with displacement of air particles, which requires some power of the engine. The expenditure of power used to overcome air resistance is composed of the following components:

(a) head resistance caused by the difference in air pressure ahead and behind the moving automobile (nearly 55-60 per cent of all air resistance);

(b) resistance created by projecting parts: running boards, mudguards, number plate (12-18 per cent);

(c) resistance arising when air passes through the radiator and the space under the bonnet (10-15 per cent);

(d) friction of the external surfaces against the neighbouring layers of air (8-10 per cent);

(e) resistance caused by the pressure difference on top and at the bottom of the automobile (5-8 per cent).

To simplify the calculations, the elemental forces of air resistance distributed over the entire surface of a motor vehicle are replaced by a concentrated *force of air resistance* P_{air}. The point

of application of the force P_{air} is called the *sail centre of a motor vehicle.*

It has been found experimentally that the force of air resistance (in N)

$$P_{air} = k_{air} F_{air} v^2 \tag{7.36}$$

where k_{air} = air resistance coefficient (streamlining coefficient) depending on the form and quality of finish of the motor vehicle surface, $N \cdot s^2/m^4$

F_{air} = head area of the motor vehicle, m^2

The streamlining coefficient k_{air} is numerically equal to the force of air resistance in N built up by one square metre of the head area of the motor vehicle as it moves at 1 m/s.

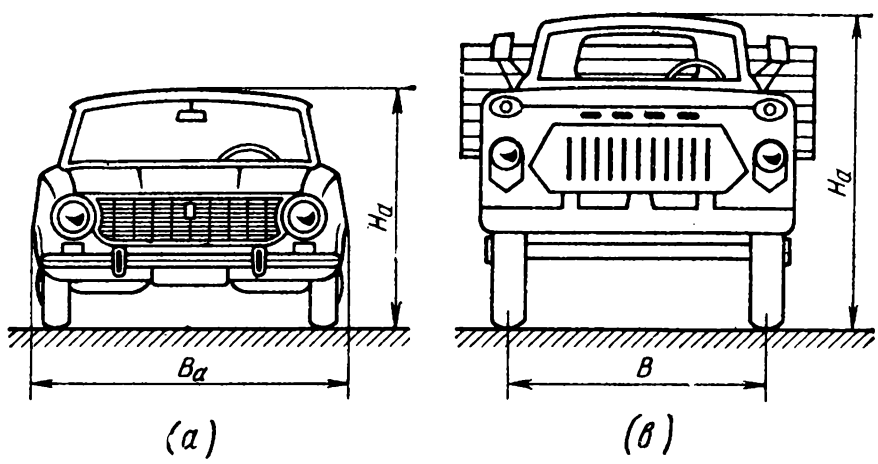


Fig. 7.8. Head areas of motor vehicles (*a*) car; (*b*) truck

The head area F_{air} of a motor vehicle is the area of its projection onto the plane perpendicular to the longitudinal axis of the motor vehicle (Fig. 7.8). It is rather difficult to find accurately the head area, for this purpose the vehicle should be measured and its external contour drawn. For this reason, the area F_{air} (in m^2) of a truck (Fig. 7.8*b*) is commonly determined by means of the approximate formula:

$$F_{air} \approx BH_a \tag{7.37}$$

where B = track in m

H_a = maximum height of the automobile in m

In the case of cars use is made of the formula (Fig. 7.8*a*):

$$F_{air} = 0.78 B_a H_a$$

where B_a is the maximum width of the automobile in m.

The product $k_{air}F_{air}$ is known as the *streamlining factor* and is denoted by the letter W_{air}.

Mean values of k_{air}, F_{air} and W_{air} are illustrated in Table 7.6.

TABLE 7.6

Parameters of Streamlined Motor Vehicles

Motor vehicles	k_{air}, N·s²/m⁴	F_{air}, m²	W_{air}, N·s²/m²
Cars:			
closed	0.20-0.35	1.6-2.8	0.3-1.0
open	0.40-0.50	1.5-2.0	0.6-1.0
Trucks	0.60-0.70	3.0-5.0	1.8-3.5
Wagon-type buses	0.25-0.40	4.5-6.5	1.1-2.6
Racing cars	0.13-0.15	1.0-1.3	0.13-0.2

The power (in kW) required to overcome air resistance

$$N_{air} = \frac{P_{air}v}{1000} = \frac{W_{air}v^3}{1000} \tag{7.38}$$

Example. Determine the force and power of air resistance when a motor vehicle ($k_{air} = 0.25$ N·s²/m⁴, $F_{air} = 2.3$ m²) moves at a speed of 10 and 20 m/s.

The force of air resistance:

at a speed of 10 m/s

$$P_{air} = k_{air}F_{air}v^2 = 0.25 \times 2.3 \times 100 = 57.5 \text{ N}$$

at a speed of 20 m/s

$$P_{air} = 0.25 \times 2.3 \times 400 = 230 \text{ N}$$

The power of air resistance:

at a speed of 10 m/s

$$N_{air} = \frac{P_{air}v}{1000} = \frac{57.5 \times 10}{1000} = 0.575 \text{ kW}$$

at a speed of 20 m/s

$$N_{air} = \frac{230 \times 20}{1000} = 4.6 \text{ kW}$$

The streamlining coefficient is estimated by means of the running-out method (see Sec. 9.5) or by blowing air onto the motor vehicle or its model in a wind tunnel. The blowing diagram is shown in Fig. 7.9*a*. A car *4* is suspended inside wind tunnel *1* which contains fan *3* with electric motor *2*. The guide lattice *6* straightens out the streams of air and eliminates turbulence at the tunnel inlet. The stream of air created by the fan tends to shift the car with the same force P_{air} that would act on the car as it moves at a speed equal to the speed of the air stream. Having determined the force P_{air} by balance *5*, and the speed of the air stream by wind gauge *7* and knowing the head area of the car,

it is possible to calculate the coefficient k_{air} using formula (7.36). When a full-sized car is used for the test, a large-size tunnel is required as well as powerful fans to build up appreciable fast moving air streams. For this reason, a sealed-down model 1/5-1/10 of actual size is used and not the car itself.

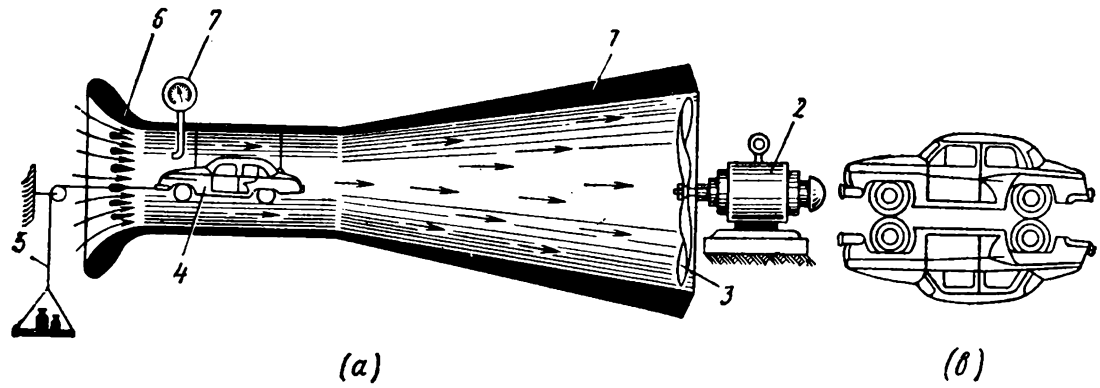


Fig. 7.9. Aerodynamic tests of a motor vehicle
(*a*) diagram of wind tunnel; (*b*) position of models in wind tunnel; *1*—wind tunnel; *2*—electric motor; *3*—fan; *4*—car; *5*—balance; *6*—guide lattice; *7*—wind gauge

The model suspended in the tunnel is subjected to streamed air on all sides, whereas on the road the car is enveloped by air mainly on the top and lateral sides. The upper layers of air between the car and the road are entrained by the car, while the bottom layers near the road surface remain immobile. Therefore,

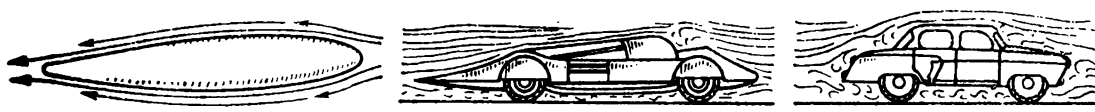


Fig. 7.10. Streamlining of an ideal body and motor vehicles of different types

the value of the coefficient k_{air} determined by blowing air onto a single model is less than the actual one. Not one but two models, one above the other, are used to eliminate this shortcoming (Fig. 7.9*b*).

To improve streamlining and reduce air resistance, the windshield of a motor vehicle is inclined and protruding parts made flush with the body. The number of projecting parts in racing

cars are reduced to the minimum and the rear of the body is elongated for streamlining.

The air resistance of trucks can be reduced if the load platform is covered by canvas stretched between the roof of the driver's cabin and the tail board, or with the aid of special cowlings which reduce the air turbulence.

Some examples of streamlining of different motor vehicles are shown in Fig. 7.10.

7.6. Equation of Motion of a Motor Vehicle

The equation of motor vehicle motion is the basic problem in traction dynamics. It interrelates the forces moving the motor vehicle with the forces of resistance to motion and makes it possible to estimate the nature of its motion at each moment of time.

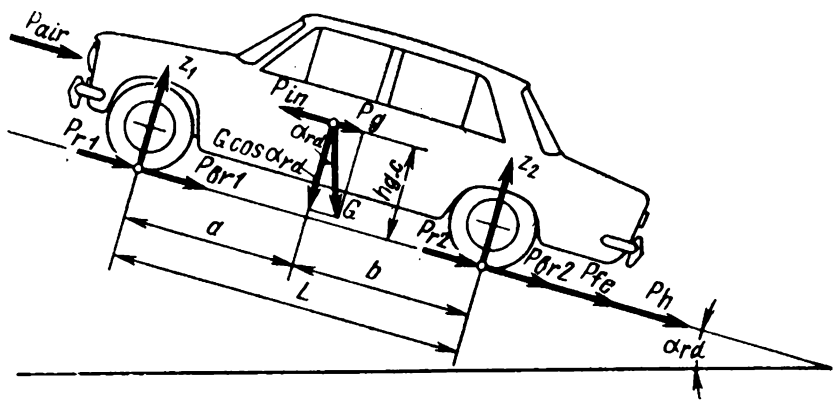


Fig. 7.11. Forces and moments acting on a car

It is assumed when studying the dynamic characteristics of a motor vehicle that its possibilities are limited only by the engine power and the cohesion between the driving wheels and the road. Other restrictions imposed, for example, by the requirements of safe traffic or comfort are disregarded. Therefore, this section deals only with linear motion. The features of curvilinear motion and its effect on the dynamic characteristics are outlined in the chapters describing the stability and steerability of motor vehicles (see Chapters XIII and XIV).

Let us consider the forces and moments acting on a motor vehicle during ascent and acceleration (Fig. 7.11). The centre of gravity of the motor vehicle is acted upon by the force of gravity $G = M_a g$ and also the force of inertia P'_{in} of the progressively moving masses directed oppositely to acceleration. The force of inertia (in N)

$$P'_{in} = M_a j \tag{7.39}$$

where M_a = mass of the motor vehicle, kg
j = acceleration, m/s^2

Applied to the wheels are the moments of rolling resistance M_{r1} and M_{r2} and also the moments of inertia M_{in1} and M_{in2}. Normal reactions Z_1 and Z_2 and tangential reactions X_1 and X_2 act on the tyres from the road. The force of air resistance P_{air} is applied to the sail centre of the motor vehicle at a height of h_{air}. In addition, a resistance force P_{tra} that a trailer offers to motion can be applied to the towing hook of the motor vehicle.

Let us project all the forces onto the surface of the road:

$$X_2 - X_1 - P'_{in} - P_g - P_{air} - P_{tra} = 0 \qquad (7.40)$$

Without a trailer

$$X_2 - X_1 - P'_{in} - P_g - P_{air} = 0 \qquad (7.41)$$

Substituting the forces X_1, X_2 and P'_{in} in formula (7.41) for their values in conformity with expressions (7.27), (7.28) and (7.39) we get

$$P_t - \frac{I_1}{r^2} j - \frac{\eta_{tr} I_f i_{tr}^2 + I_2}{r^2} j - M_a j - f(Z_1 + Z_2) - P_g - P_{air} = 0 \qquad (7.42)$$

At the same time

$$f(Z_1 + Z_2) + P_g = P_r + P_g = P_{rd}$$

Hence,

$$P_t - \left(1 + \frac{\eta_{tr} I_f i_{tr}^2 + I_r}{M'_a r^2}\right) M_a j - P_{rd} - P_{air} = 0 \qquad (7.43)$$

In the latter expression, I_r is the total inertia moment for all wheels in $N \cdot m \cdot s^2$.

The second term of equation (7.43) is the force necessary to impart to a motor vehicle an acceleration equal to j. Comparing this acceleration with the force P'_{in}, it can clearly be seen that the expression in the brackets shows by how many times the energy expended to accelerate the rotating and progressively moving parts of a motor vehicle is larger than the energy needed to accelerate a vehicle whose parts move only translationally. Thus, this expression takes into account the effect of the rotating masses and is therefore called the *coefficient accounting for rotating masses* δ_{rot}, and the second term — the *reduced force of inertia* P_{in} of a motor vehicle:

$$\delta_{rot} = 1 + \frac{\eta_{tr} I_f i_{tr}^2 + I_r}{M_a r^2} \qquad (7.44)$$

$$P_{in} = \delta_{rot} P'_{in} = \left(1 + \frac{\eta_{tr} I_f i_{tr}^2 + I_r}{M_a r^2}\right) M_a j \qquad (7.45)$$

The energy expended to accelerate the rotating parts of an engine at direct drive is 2-3 times and at lower gears 8-10 times larger than the energy expended to accelerate the wheels.

If the accurate value of the moments of inertia I_f and I_r is unknown the coefficient δ_{rot} can be found from the empirical formula

$$\delta_{rot} = 1 + (\delta_1 + \delta_2 i_r^2) \frac{G_a}{G} \tag{7.46}$$

where G_a = weight of the motor vehicle with full load, N
G = weight of the motor vehicle with a given load, N
$\delta_1 \approx \delta_2 \approx 0.03\text{-}0.05$

If a motor vehicle moves under inertia, i.e., with the engine disconnected from the transmission, the coefficient accounting for rotating masses is denoted by the letter δ_{in} and determined by the formula

$$\delta_{in} = 1 + \frac{I_r}{M_a r^2} \approx 1 + 0.05 \frac{G_a}{G} \tag{7.47}$$

Introducing the coefficient δ_{rot} into equation (7.43) the sought-for equation can be obtained for the general case of motor vehicle motion

$$P_t - P_{in} - P_r - P_g - P_{atr} = P_t - P_{in} - P_{rd} - P_{atr} = 0 \tag{7.48}$$

In equation (7.48) the values of the forces P_{in} and P_g should be inserted with the signs which depend on the nature of motion.

Note should be made of the arbitrary assumptions made in deriving the latter equation. Firstly, the force P_t refers to steady motion and the equation is derived for the general case of motor vehicle motion. Secondly, the force P_r is regarded as an external force resisting the motion, which represents it as the component of the force P_{rd}. In actual fact, when a motor vehicle runs on a road with a hard surface the principal portion of the energy expended on rolling is used to overcome the internal friction in the tyres. For this reason the force P_r may be called external only arbitrarily.

7.7. Tractive Force According to Conditions of Tyre Road Grip

It has been assumed when estimating the tractive force that its magnitude depends only on the parameters of a motor vehicle (M_e, i_{tr}, η_{tr}, r). This does not mean, however, that by increasing, for example, the gear ratio of transmission it is possible to realize any large tractive force, because its limiting value is restricted by the grip of tyres with the road surface.

In the absence of the transverse forces, the magnitude of the gripping force

$$P_{gr} = \varphi_x Z \geqslant X$$

In the case of uniform rolling of a wheel

$$X = P_t - fZ; \quad P_t \leqslant (\varphi_x + f) Z$$

The value of the coefficient φ_x is usually much higher than the coefficient f and, therefore, the condition of wheel rolling without slip may with a small error be represented as

$$P_t \leqslant P_{gr} = \varphi_x Z \tag{7.49}$$

If the tractive force is less than the gripping force P_{gr}, the driving wheel rolls without skidding. If the tractive force exceeds the gripping force, the driving wheel slips and only part of the force equal to $\varphi_x Z$ is employed for motion. The rest of the force P_t causes an accelerated rotation of the wheels, which continues until the power expended on slipping equalizes the excess of power delivered to the wheels. Slipping is most frequently observed when a motor vehicle is started sharply and when it overcomes a high resistance to motion on a slippery road.

When the grip coefficient is estimated experimentally, a truck tractor tows, by means of a wire rope with a built-in dynamometer, a bogie whose wheels are braked. After measuring the force P_x necessary for uniform motion with a dynamometer and knowing the weight G_b of the bogie the grip coefficient can be found from the formula $\varphi_x = \frac{P_x}{G_t}$.

On roads with hard surface the grip coefficient primarily depends on sliding friction between the tyre and the pavement. On soft deformable roads the grip coefficient first depends on the resistance of the soil to shear and the internal friction in the soil. As they plunge into the soil, the projections (cleats) on the tread of the driving wheel deform and compact the soil increasing thereby the resistance to shear to a certain limit. However, the soil is then destroyed and the grip coefficient drops. If a hard surface is wettened the grip coefficient at first sharply drops because of the film formed by the water and soil particles which tend to reduce friction between the tyre and the road. The drop in the grip coefficient is quite appreciable when a film of clay forms on the surface. Heavy rain can wash off this film and as a result the value of φ_x approximates values typical of dry hard surface. The value of φ_x is considerably affected by the pattern of the tread. The tyre tread of cars has a fine pattern (Fig. 7.12*a*, *b*) ensuring good cohesion with a hard surface. A large pattern with wide and deep recesses is typical of the tyre tread of trucks

(Fig. 7.12 *c-h*). Such tyres plunge into the soil and improve the cross-country ability.

The tread wear impairs the tyre road grip. The minimum grip coefficient has the tyres with completely worn-out tread pattern. The use of such tyres is prohibited.

An insufficient value of the coefficient φ_x is the cause of many road accidents. This coefficient should not be less than 0.4 for safe traffic.

If slipping (or skidding) of the tyre the grip coefficient rises and reaches its maximum at 20-25 per cent slip value. With full

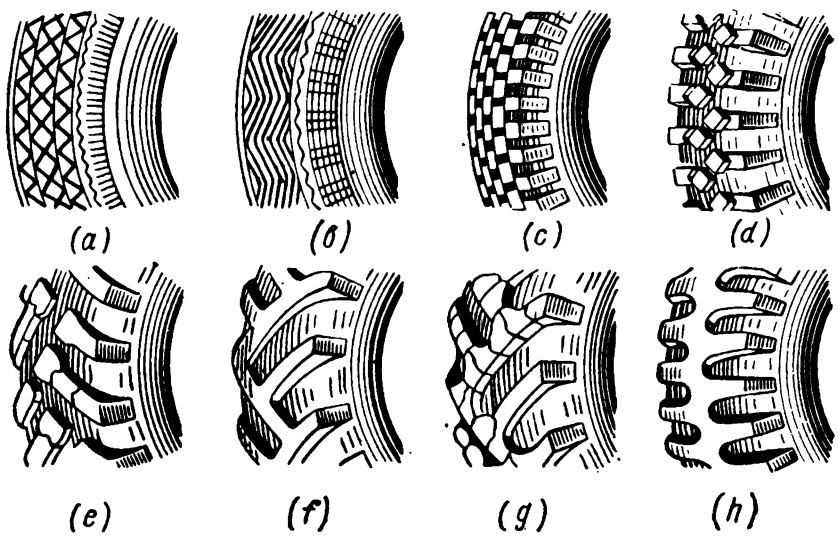


Fig. 7.12. Patterns of tyre treads (*a*), (*b*) cars, (*c*)-(*h*) trucks

slipping of the driving wheels or skidding of the braking wheels the value of φ_x may be 10-25 per cent less than the maximum.

With a growth in the speed the coefficient φ_x first slightly increases reaching its maximum on a hard dry surface at a speed of 3-4 m/s, it then diminishes. At a speed of 40 m/s the φ_x value may be several times less than that at the speed of 10 m/s.

TABLE 7.7

Grip Coefficient φ

Pavement	State of surface	
	dry	wet
Asphalt-concrete, cement-concrete	0.7-0.8	0.35-0.45
Broken stone	0.6-0.7	0.3-0.4
Dirt road	0.5-0.6	0.2-0.4
Icy road	0.1-0.2	
Snow-covered road	0.2-0.3	

Since many factors affect the φ_x value, which are difficult to account for, the calculations are usually done using the mean values of the coefficient φ_x (Table 7.7). The coefficient of transverse grip φ_y depends on the same factors and is usually assumed as equal to the coefficient φ_x.

Conditions for Driving

In conformity with equation (7.43) a prolonged incessant motion of a motor vehicle is possible only if

$$P_t \geqslant P_{rd} + P_{air} \tag{7.50}$$

Inequality (7.50) interrelates the design parameters of a motor vehicle with the external resistances to motion. It should be satisfied but this is not enough for continuous motion as it is possible only if the driving wheels do not skid. Taking into account formula (7.49) the condition of incessant motion can be expressed as follows:

$$\varphi_x Z_2 \geqslant P_t \geqslant P_{rd} + P_{air} \tag{7.51}$$

Thus, continuous motion without stalling is possible only if the tractive force on the driving wheels does not exceed the gripping force, but it must be larger than or, at least, equal to the sum of the forces of the road and air resistance.

7.8. Normal Reactions of the Road

Let us determine the normal reactions acting on the wheels of a motor vehicle standing on a horizontal road. The weight of the motor vehicle is assumed to act on the centre of gravity at distance a from the front axle and at distance b from the rear one (see Fig. 7.11).

When a motor vehicle is static, the normal reactions of the road acting on the front (Z_1) and rear (Z_2) wheels are equal to the components G_1 and G_2 of the vehicle weight, respectively:

$$Z_1 = G_1 = G\frac{b}{L}; \quad Z_2 = G_2 = G\frac{a}{L} \tag{7.52}$$

where L is the distance between the axles in m.

As the motor vehicle moves, the road normal reactions never remain constant but change under the action of the forces and moments applied to the motor vehicle.

Disregarding the elasticity of springs it may be assumed that the body of the motor vehicle is rigidly linked with the front and rear axles. However, the wheels are hinge-jointed to the axles by means of bearings. Therefore, in motion, the body may perform angular displacements with respect to one of the axles. The body is turned under the action of moments such as the moment transmitted from the engine to the wheels, the moments of the wheel

inertia forces arising in nonuniform motion, the moments of the forces of rolling resistance, the moment created by any force whose line of action does not pass through the front and rear axles.

Let us now determine the normal reactions acting on the wheels of a two-axle motor vehicle as it accelerates uphill (see Fig. 7.11). For this purpose we write the equations of moments of all forces with respect to the rear and front axles. For simplicity it will be assumed that the height of the sail centre h_{air} is equal to the height of the gravity centre $h_{g.c}$ and that the moments of the forces of inertia of the wheels M_{in1} and M_{in2} may be neglected. In this case

$$Z_1 L + (P_g + P_{air} + P'_{in})(h_{g.c} - r) + M_r + \\ + P_{tra}(h_{tra} - r) + (X_2 - X_1) r - G \cos \alpha_{rd} b = 0$$

$$Z_2 L - (P_g + P_{air} + P'_{in})(h_{g.c} - r) - M_r - P_{tra}(h_{tra} - r) - \\ - (X_2 - X_1) r - G \cos \alpha_{rd} a = 0$$

where $M_r = M_{r1} + M_{r2}$ in N·m

h_{tra} = height of the trailer coupling device, m

The equations obtain give us the values of the normal reactions Z_1 and Z_2 (in N):

$$Z_1 = \frac{G \cos \alpha_{rd} b}{L} - \frac{(P_g + P_{air} + P'_{in})(h_{g.c} - r) + M_r + (X_2 - X_1) r}{L} - \\ - \frac{P_{tra}(h_{tra} - r)}{L} \quad (7.53)$$

$$Z_2 = \frac{G \cos \alpha_{rd} a}{L} + \frac{(P_g + P_{air} + P'_{in})(h_{g.c} - r) + (X_2 - X_1) r}{L} + \\ + \frac{P_{tra}(h_{tra} - r)}{L} \quad (7.54)$$

Since the moment $M_r = f(Z_1 + Z_2) r = P_r r$ and the sum of the moments of inertia of the wheels $I_1 + I_2 = I_r$, then accounting for expressions (7.27), (7.28) and (7.39) it may be written that

$$Z_1 = \frac{G \cos \alpha_{rd} b}{L} - \frac{(P_g + P_{air} + M_{aj})(h_{g.c} - r) + \left(P_f - \frac{\eta_{tr} I_f i_{tr}^2 + I_r}{r^2}\right) r}{L} - \\ - \frac{P_{tra}(h_{tra} - r)}{L} \quad (7.55)$$

$$Z_2 = \frac{G \cos \alpha_{rd} a}{L} + \frac{(P_g + P_{air} + M_{aj})(h_{g.c} - r) + \left(P_f - \frac{\eta_{tr} I_f i_{tr}^2 + I_r}{r^2}\right) r}{L} + \\ + \frac{P_{tra}(h_{tra} - r)}{L} \quad (7.56)$$

It follows from these equations that, when a motor vehicle moves, the normal reactions of the road differ from the reactions

acting on the wheels of a stationary vehicle. The reaction on the front wheels diminishes and on the rear wheels grows with an increase in the gradient, intensity of acceleration and the forces of resistance P_r, P_{air} and R_{tra}.

Let us introduce a reaction change coefficient in order to evaluate the number of times the reactions Z_1 and Z_2 change during motion as compared with the vertical loads G_1 and G_2 with a motor vehicle in the static condition.

The reaction change coefficient m_r is the ratio of the normal reaction acting on the axle of a moving vehicle to the reaction which acts on the same axle when the motor vehicle stands on a horizontal road:

$$m_{r1} = Z_1 : G_1; \quad m_{r2} = Z_2 : G_2$$

As a motor vehicle is accelerated, the maximum values of the reaction change coefficients lie within: $m_{r1} = 0.65\text{-}0.7$ for the front axle and $m_{r2} = 1.2\text{-}1.35$ for the rear axle.

These values testify to the fact that during acceleration the load on the front axle of the motor vehicle diminishes, and that on the rear axle increases as compared with the loads when the vehicle is in static condition. The reverse occurs when a motor vehicle is braked. This explains the rise of the front portion of the motor vehicle as it picks up speed and its lowering during braking.

Example. Find the reaction change coefficients for a car moving uniformly uphill ($\alpha_{rd} = 3°30'$) at a speed of 20 m/s if $a = 1.46$ m, $b = 1.34$ m, $L = 2.8$ m, $r = 0.33$ m, $h_{g.c} = 0.6$ m. The values of P_r, P_{air} and P_g are to be taken from the previous examples.

Reactions with a motor vehicle in static condition:

$$Z_1 = \frac{Gb}{L} = \frac{17.9 \times 1.34}{2.8} = 8.56 \text{ kN}$$

$$Z_2 = \frac{Ga}{L} = \frac{17.9 \times 1.46}{2.8} = 9.34 \text{ kN}$$

Reactions when the vehicle moves at $v = 20$ m/s ($j = 0$) with the force $P_t = P_r + P_g + P_{air}$:

$$Z_1 = \frac{G \cos \alpha_{rd} b}{L} - \frac{(P_g + P_{air}) h_{g.c} + P_r r}{L} = \frac{17.9 \times 0.998 \times 1.34}{2.8} -$$

$$- \frac{(1.07 + 0.23) \times 0.6 + 0.318 \times 0.33}{2.8} = 8.55 - 0.317 \approx 8.23 \text{ kN}$$

$$Z_2 = \frac{G \cos \alpha_{rd} a}{L} + \frac{(P_g + P_{air}) h_{g.c} + P_r r}{L} =$$

$$= \frac{17.9 \times 0.998 \times 1.46}{2.8} + 0.317 = 9.32 + 0.317 \approx 9.64 \text{ kN}$$

The reaction change coefficients:

$$m_{r1} = \frac{8.23}{8.56} = 0.96; \quad m_{r2} = \frac{9.64}{9.34} = 1.03$$

Chapter VIII TRACTION DYNAMICS OF A MOTOR VEHICLE

8.1. Force Balance

The differential equation for the motion of a motor vehicle has been obtained above. It is impossible to solve this equation in the general form, because the exact functional dependences relating the basic acting forces P_t, P_{tr} and P_r to the speed of the motor vehicle are unknown. For this reason the equations of motion are usually solved approximately by using simple and visual graphical and analytical methods which yield, however, the required accuracy of the results. The most popular are *the force balance method, the power balance method* and *the dynamic characteristic method.* Let us begin with the force balance method.

Rewriting equation (7.48) as

$$P_t = P_r + P_g + P_{air} + P_{in} = P_{rd} + P_{air} + P_{in} \qquad (8.1)$$

we can obtain the *force balance equation* which can be solved graphically as shown in Fig. 8.1*a*.

The first thing is to plot the motor vehicle traction characteristic (see Sec. 7.3). The curve P_{rd} in the bottom portion of the graph is plotted for one value of the coefficient ψ. The curve of the air resistance force P_{air} is constructed by plotting the values of this force upwards from the corresponding values of the force P_{rd}.

The curve of the total resistance $P_{rd} + P_{air}$ determines the magnitude of the tractive force needed to run the motor vehicle at constant speed. If at a certain speed the curve P_t passes above the curve $P_{rd} + P_{air}$ the section P_{res} enclosed between these curves represents the unused part ("reserve") of the force which can be utilized to overcome increased road resistance or to accelerate the vehicle.

The obtained force balance graph can be used to estimate the main dynamic characteristics of a motor vehicle in uniform motion. For instance, the maximum speed v_{max} is determined by the abscissa at the intersection point of curves P_t and $P_{rd} + P_{air}$, because in this case the force reserve and, therefore, the acceleration are zero. If the curve P_t lies below the total curve, the motor vehicle slows down.

To evaluate the maximum road resistance which can be overcome by the motor vehicle in uniform motion, for example, at a speed of v_1, the section *ab* equal to the force P_{air} in the selected

scale at a speed of v_1 should be subtracted from the ordinate P_t. The sum of the sections P_{res} and bc represents in the same scale the force $P_{rd\ max}$.

Slipping of the driving wheels is allowed for by determining the grip force $P_{gr} \approx \varphi_x G_2$ for a given value of φ_x and by drawing the horizontal line as shown in Fig. 8.1*a*. The condition $P_t \leqslant P_{gr}$ is observed in the area below this line and is not observed in the area above it, and prolonged motion is impossible

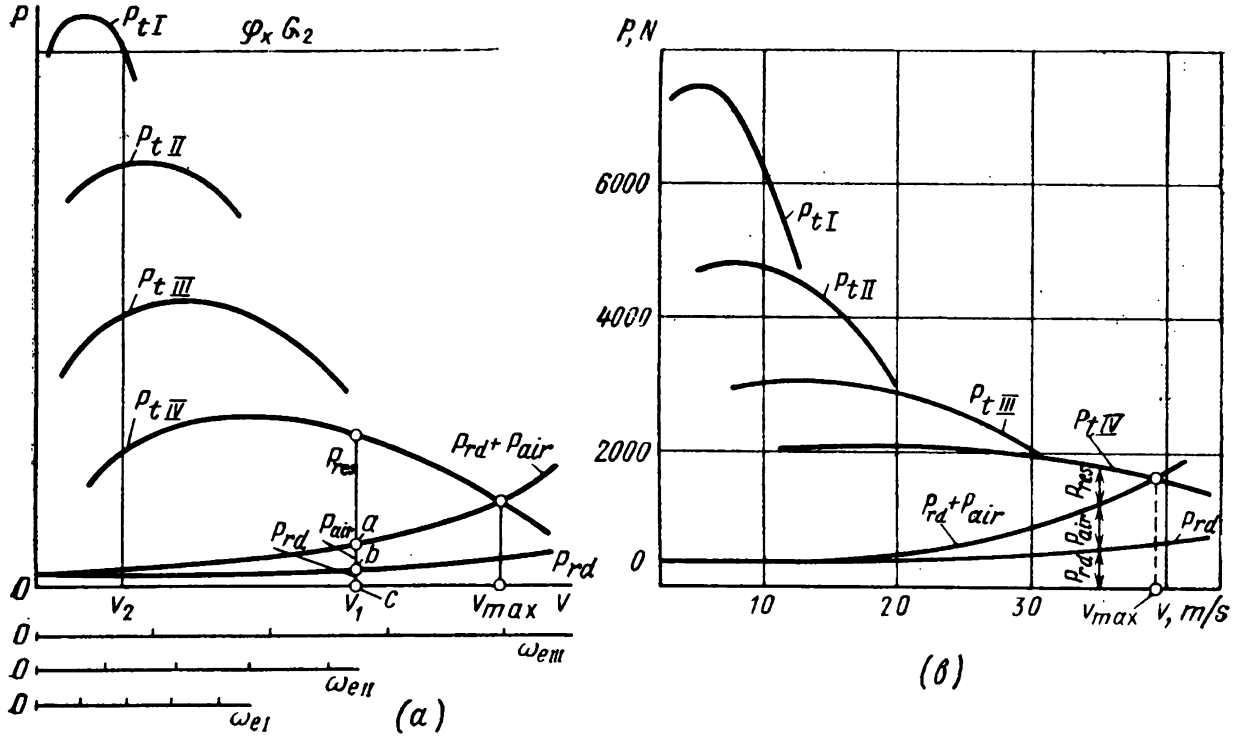


Fig. 8.1. Force balance of a motor vehicle
(*a*) general view; (*b*) force balance of a motor vehicle accepted for calculation

in this area. Therefore, incessant motion in this example in fourth, third and second gears can be effected within the entire range of speeds from the minimum to the maximum one. In first gear and full engine load the wheels can move without skidding only at a speed larger than or equal to the v_2 speed. For motion in first gear with a speed less than v_2 it is necessary to slightly shut the throttle and reduce thereby the force P_t.

The force balance of a motor vehicle accepted for calculation, plotted as shown above, is illustrated in Fig. 8.1*b*. According to this graph, the maximum rated speed in motion on an asphalt road is equal to 39.5 m/s.

In the force balance equation written for the general case of motion, all the resistance forces are conditionally given a plus sign. However, the forces P_g and P_{in} may be either resistance or motive forces depending on the nature of the motion. In uphill

motion ($+\alpha$) and also during acceleration of a motor vehicle ($+j$) the values of these forces should be inserted into equation (8.1) with plus signs, and in the case of downhill motion ($-\alpha$) or deceleration ($-j$), with minus signs.

8.2. Power Balance

The dynamic properties of a motor vehicle can be analyzed by comparing the tractive power N_t with the power required to overcome the resistance to motion instead of using the ratio of the forces. By analogy with the equation of force balance (8.1), the equation of power balance may be written in the following form:

$$N_t = N_e - N_{tr} = N_r + N_g + N_{air} + N_{in} \qquad (8.2)$$

where $N_{in} = \frac{P_{in} v}{1000}$ is the power expended to overcome the inertia of a motor vehicle in kW.

In a developed form and with regard to the above formulas

$$N_t = \frac{fG \cos \alpha_{rd} v}{1000} + \frac{G \sin \alpha_{rd} v}{1000} + \frac{W_{air} v^3}{1000} + \frac{\delta_{rot} M_a j v}{1000} \qquad (8.3)$$

First we consider the solution of equation (8.3) for the case of motion in one gear. Let us plot the speed characteristic of power N_e in the N-v coordinates as shown in Fig. 8.2*a*. We now plot the values of the power N_{tr}, used to overcome friction in the transmission, downwards from the curve N_e to obtain curve N_t. If the value of N_{tr} is unknown, the power N_t can be found from the formula $N_t = \eta_{tr} N_e$.

The graph $N_{rd} = f(v)$ is plotted at the bottom by determining N_{rd} from formula (7.35). If it is assumed that the coefficient f is constant, this graph takes the form of an inclined straight line passing through the origin of the coordinates. For a speed larger than 14-16 m/s account should be taken of the dependence of the coefficient f on the speed, this coefficient being estimated, for example, from formula (7.31). The values of the air resistance power N_{air} are plotted upwards from the curve N_{rd} (see the curve $N_{rd} + N_{air}$). The sections of the ordinates between the curve $N_{rd} + N_{air}$ and the X-axis represent the total power expended on overcoming the road and air resistance. The sections N_{res} between the curves N_t and $N_{rd} + N_{air}$ form the power reserve which can be expended to overcome the increased road resistance or to accelerate the motor vehicle. During uniform motion, the power N_t is used only to overcome the road and air resistance. If the throttle is fully open the motor vehicle accelerates to the maximum speed v_{max} when the power N_t is equal to the sum of the powers N_{rd} and N_{air} (point A in Fig. 8.2*a*). The throttle should

be slightly closed for uniform motion on the same road at a speed less than v_{max}. For instance, for motion at v_1 the vehicle should be throttled down so that the power N_t changes according to the curve shown by the dash line in Fig. 8.2*a*.

The power balance graph is constructed for all the transmission gear ratios in the same manner, but several pairs of curves N_e and N_{tr} are plotted instead of one depending on the number of

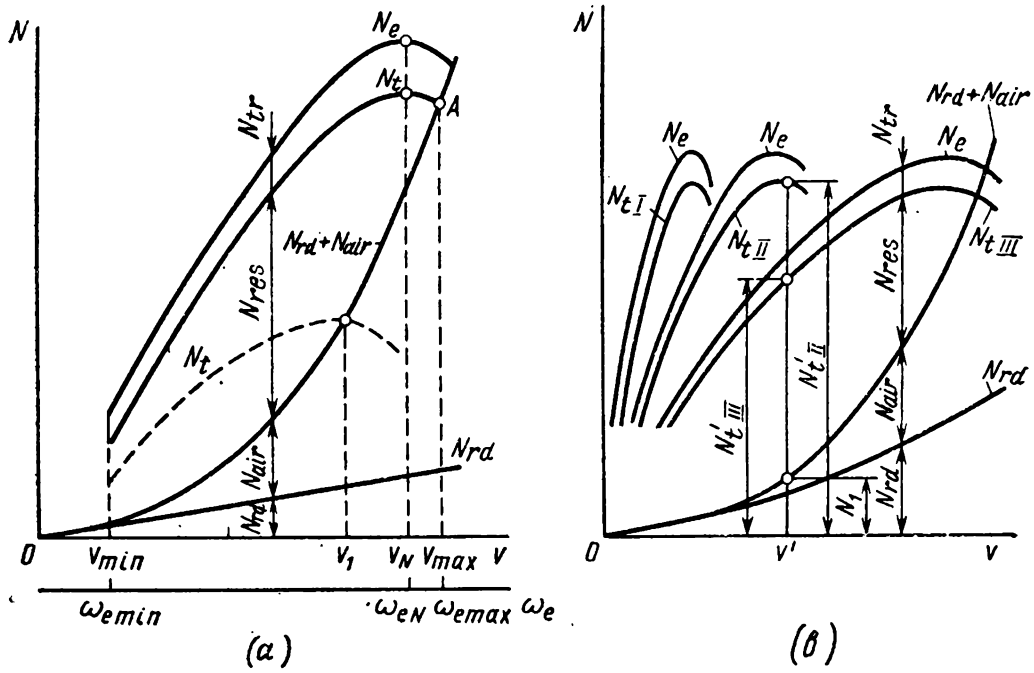


Fig. 8.2. Power balance of a motor vehicle (*a*) in highest gear; (*b*) in all gears

steps in the transmission gearbox (see Fig. 8.2*b*). A change in the gear ratio i_g alters only the speed of the motor vehicle, and the powers N_e and N_{tr} remain as before if the changes in the transmission efficiency through the gears are neglected.

During calculations the powers N_g and N_{in} in ascent and also when the motor vehicle accelerates should be inserted into the power balance equation written for the general case of motion with plus signs, and with minus signs if the motor vehicle travels downhill or decelerates.

The degree of engine power utilization U is the ratio of the power necessary to move a motor vehicle to the power that can be developed by the engine with the throttle fully open

$$N = \frac{N_{rd} + N_{air} + N_{tr} + N_{in}}{N_e} = \frac{N_{rd} + N_{air} + N_{in}}{\eta_{tr} N_e} = \frac{N_{rd} + N_{air} + N_{in}}{N_t} \quad (8.4)$$

If a motor vehicle moves uniformly

$$U = \frac{N_{rd} + N_{air}}{\eta_{tr} N_e} = \frac{N_{rd} + N_{air}}{N_t} \tag{8.5}$$

The degree of power utilization depends on the type and state of the road, the speed and gear ratio of the transmission i_{tr}. The better the road and the less the ψ coefficient, the slower the speed, and the higher the gear ratio i_{tr}, less use will be made of the engine power.

For instance, if the speed is v' (Fig. 8.2*b*) the total power of the road and air resistance is N_1 and the power delivered to the driving wheels with the full-open throttle will be $N'_{t\,III}$ at direct drive and $N'_{t\,II}$ in second gear. Then, the degree of power utilization with direct drive will be $U_1 = \frac{N_1}{N'_{t\,III}}$ and in second gear $U_2 = \frac{N_1}{N'_{t\,II}}$. But $N'_{t\,II} > N'_{t\,III}$ and therefore $U_2 < U_1$.

Chapter IX DYNAMIC CERTIFICATE OF A MOTOR VEHICLE

9.1. Dynamic Factor

It is rather difficult to employ in practice the methods of the force and power balance, since several P_{rd} and $P_{rd} + P_{air}$ curves have to be plotted on the force balance graph for various values of the road resistance coefficient and a number of N_{rd} and $N_{rd} + N_{air}$ curves, on the power balance graph. This makes the graphs complicated and requires more time. In addition, the power and force balances cannot be used to compare the dynamic characteristics of motor vehicles of different weights, because when they move in identical conditions the force and power needed to overcome the road resistance are different too. The method of solving the equation of motion of a motor vehicle with the aid of the dynamic characteristic suggested by E. A. Chudakov is free from these shortcomings.

The dynamic factor D of a motor vehicle is the ratio of the difference in the tractive force and the force of air resistance to the weight of the motor vehicle:

$$D = \frac{P_t - P_{air}}{G} = \left(\eta_{tr} \frac{M_e i_{tr}}{r} - W_{air} v^2\right) \frac{1}{G} \tag{9.1}$$

The value of D depends only on the motor vehicle design parameters and can, therefore, be determined for each particular model. The dynamic factor is larger for low gears than for high gears due to the increase in the force P_t and the drop in the force P_{air}. Since the force P_{air} is small when a motor vehicle moves at a low speed and may be neglected, the dynamic factor in low gears for speeds up to 4-5 m/s can be found from the approximate formula

$$D = \frac{P_t}{G}$$

In order to correlate the dynamic factor with the conditions of motion, let us transfer the force P_{air} in equation (8.1) to the left side and divide both sides by the motor vehicle weight G. As a result, we obtain

$$D = \psi + \frac{\delta_{rot}}{g} j \tag{9.2}$$

In uniform motion acceleration and deceleration are equal to zero and, therefore, the value of D will also determine the value of ψ. Thus, the value of the dynamic factor D_v with the maximum speed evaluates the road resistance ψ_v which can be overcome by the motor vehicle at this speed.

The maximum dynamic factor D_{max} determines the value of the maximum road resistance ψ_{max} overcome in uniform motion in first gear. The values of v_{max}, D_{max} and D_v are the principal dynamic characteristics of a motor vehicle in uniform motion.

In conformity with expression (9.2), the following condition should be observed for prolonged continuous motion of a motor vehicle:

$$D \geqslant \psi \tag{9.3}$$

Let us find the limiting value of the tractive force (in N) by the conditions of grip, to take into account also the restriction of motion due to skidding of the driving wheels:

$$P_t = P_{gr} = m_2 \varphi_x G_2$$

Inserting the found value of P_{gr} into the formula which determines the value of D we get an expression for the dynamic factor concerning grip

$$D_{gr} = \frac{D_{gr} - P_{air}}{G} = \frac{m_2 \varphi_x G_2 - P_{air}}{G} \tag{9.4}$$

If the driving wheels skid, the speed of the motor vehicle is low and it may be assumed that the force $P_{air} \approx 0$.

Then, taking $m_2 \approx 1$ we obtain

$$D_{gr} = \varphi_x \frac{G_2}{G} \tag{9.5}$$

Obviously, a motor vehicle can move without the driving wheels skidding if the following condition is observed:

$$D_{gr} \geqslant D \tag{9.6}$$

With regard to formula (9.3) the following condition for the motion of a motor vehicle can be obtained:

$$D_{gr} \geqslant D \geqslant \psi \tag{9.7}$$

Example 1. Estimate the dynamic factor for a car under full load, running with direct drive at a speed of 17.7 m/s, if the tractive force is equal to 2120 N (from the previous examples).

According to the traction conditions, the dynamic factor

$$D = \frac{P_t - P_{air}}{G} = \frac{2\,120 - 0.25 \times 2.3 \times 17 \cdot 7^2}{17{,}900} = 0.109$$

Example 2. Find the possibility of motion of the same car under full load at a speed of 17.7 m/s on a road described by the coefficients $\psi = 0.08$ and $\varphi_x = 0.1$.

The dynamic factor D_{gr} as to grip

$$D_{gr} = \varphi_x \frac{G_2}{G_a} = 0.1 \times \frac{9\,340}{17{,}900} = 0.052$$

Since

$$D = 0.109 > D_{gr} = 0.052$$

prolonged motion of the car at this speed is impossible because the driving wheels skid.

The dynamic characteristic of a motor vehicle is the graph of dependence of the dynamic factor D_a of an automobile under full load on the speed of its motion in different gears. A tentative

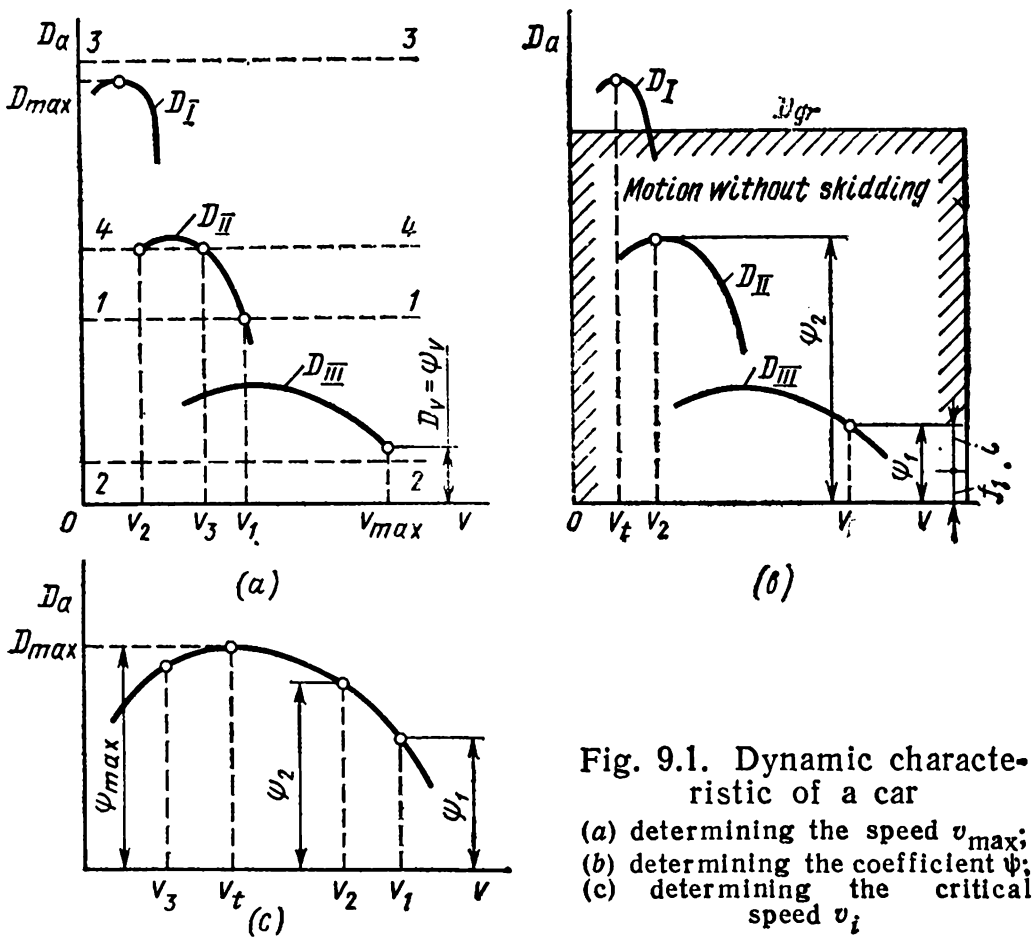


Fig. 9.1. Dynamic characteristic of a car
(*a*) determining the speed v_{max};
(*b*) determining the coefficient ψ;
(*c*) determining the critical speed v_i

dynamic characteristic of a car is shown in Fig. 9.1. The car motion equation can be solved more simply by the dynamic characteristic method than by the force or power balance methods. In this case the values of the dynamic factors are compared by the conditions of traction and grip with the value of the coefficient ψ. For instance, in order to determine the maximum speed of the car on the section of the road described by the coefficient ψ, its value should be plotted on the axis of coordinates of the dynamic characteristic to the same scale as the scale of D_a and a straight line drawn parallel to the X-axis. Several cases are possible.

(a) If the line ψ (*1-1* in Fig. 9.1*a*) intersects the curve of the dynamic factor, the maximum speed is equal to v_1, since with this speed the condition $D_a = \psi$ is observed.

(b) If the curve D_a passes above the line ψ (line *2-2*), uniform motion is impossible with full delivery of the mixture or fuel to the engine cylinders, because even in the highest gear the dynamic factor within the entire range of speeds is larger than the coefficient ψ. The motor vehicle is accelerated with an excess dynamic factor. To ensure uniform motion, the throttle should be closed slightly thereby diminishing the engine torque. In this case the maximum speed of the motor vehicle will be limited by the maximum permissible angular velocity of the engine shaft $\omega_{e\,max}$.

(c) Line *3-3* corresponds to the case when $D_a < \psi$. With this value of the coefficient ψ, motion at a constant speed is impossible and the motor vehicle can move only at low speed.

(d) If the line ψ intersects the curve D_a at two points (line *4-4*), the motor vehicle can move uniformly with the full-open throttle both at a speed of v_2 and at a speed of v_3. For uniform motion at a speed higher than v_2 and lower than v_3 the engine power should be reduced by shutting the throttle.

The dynamic characteristic can be used to solve easily the reverse problem, i.e., to determine the coefficient of road resistance offered to a motor vehicle running at a preset speed. For this purpose, the dynamic characteristic is employed to find the value of D_a at a given speed and thus estimate the value of ψ. For instance, the coefficient of road resistance is equal to ψ_1 at a speed of v_1 (Fig. 9.1*b*) and to ψ_2 at a speed of v_2.

If the value of the coefficient f is known the found value of ψ can be employed to estimate from formula (7.33) the maximum angle of ascent which can be overcome by the motor vehicle. Thus, if the coefficient of rolling resistance is equal to f_1, a motor vehicle moving at a speed of v_1 can climb up a hill whose gradient i is equal to $\psi_1 - f_1$.

It is likewise possible to find the speed and the corresponding maximum coefficient of the road resistance ψ_{max} which can be overcome by a motor vehicle in a certain gear. Obviously, the maximum coefficient ψ occurs when the value of D_a is the greatest. Therefore, a straight line parallel to the X-axis and tangential to the curve of the dynamic factor in a given gear should be drawn to determine the value of ψ (Fig. 9.1*c*). The portion cut off by this straight line on the coordinate axis represents the value of D_{max} and, therefore, ψ_{max}.

The abscissa of the point of contact will show the motor vehicle speed v_t. This speed is of prime importance for the following considerations. Let a motor vehicle be moving uniformly at a speed of v_1 on a road characterized by the coefficient ψ_1 (see

Fig. 9.1*c*). If the road resistance increases to ψ_2 the speed of the vehicle begins to diminish and the dynamic factor grow until it reaches the value of $D_2 = \psi_2$. In this case the motor vehicle will begin to run uniformly but now at a speed of v_2. If then the additional resistance disappears the dynamic factor will prove larger than the coefficient of road resistance, the speed will increase and the dynamic factor diminish until the equality $D_a = \psi$ is realized, after which the speed will no longer increase. Thus, during motion at a speed higher than v_t the steady nature of the vehicle motion will be maintained automatically despite certain changes in the coefficient ψ. At a speed equal to or smaller than v_t, for example at v_3, the additional resistance will reduce the speed and the value of D_a. A drop in D_a will in turn decrease the speed and, therefore, so that the motor vehicle may not stop, the driver should engage a lower gear, thereby increasing the value of the dynamic factor and preserving the stability of motion.

Hence, the speed v_t determines the region of the steady motion of a motor vehicle. At $v > v_t$ the motion is steady and at $v \leqslant v_t$, unsteady. For this reason the speed v_t is called the *critical speed with respect to traction conditions.*

9.2. Dynamic Certificate

The dynamic certificate of a motor vehicle is a combination of the dynamic characteristic, load nomogram and skidding control graph. The certificate makes it possible to solve the equation of motion with account of a large number of factors: design parameters of a motor vehicle (M_e, etc.), basic characteristics of the road (coefficients ψ, φ_x) and load on the vehicle.

The dynamic characteristic is plotted for a motor vehicle under full load. When the weight changes from G_a to G, the value of the dynamic factor changes and becomes equal to

$$D = \frac{P_t - P_{air}}{G} = D_a \frac{G_a}{G} \tag{9.8}$$

So as not to recalculate the value of D upon each change in the load, the dynamic characteristic is supplemented with a *load nomogram* which is constructed as follows. The X-axis of the dynamic characteristic is continued leftwards and a section of arbitrary length is plotted on it. A load scale in per cent (for trucks) or by the number of passengers (for cars and buses) is marked on this section. A straight line is drawn through the zero point of the load scale parallel to the axis D_a and the scale of the dynamic factor D_0 is marked on this line for a motor vehicle without load. The magnitude of the scale a_0 for the scale D_0 can

be determined from the formula

$$a_0 = a_a \frac{G_0}{G_a}$$

where a_a = scale of the dynamic factor for an automobile with full load

G_0 = own weight of the automobile in an equipped state including the weight of the driver, N

The equivalent divisions on the scales D_0 and D_a (for example, 0.05, 0.1, etc.) are connected by straight lines.

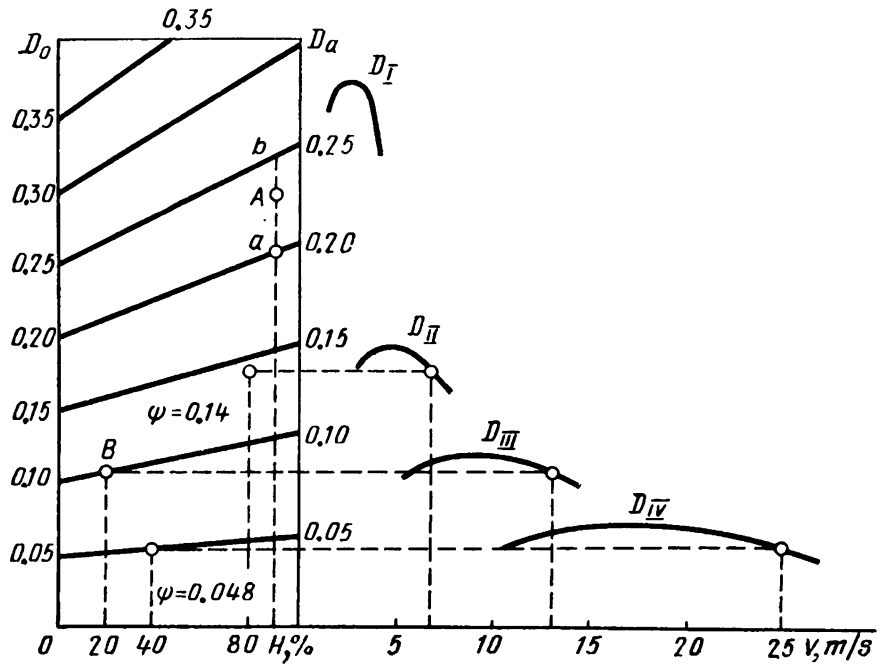


Fig. 9.2. Dynamic characteristic of a motor vehicle with a load nomogram

The inclined lines are usually drawn on the load nomogram through the rounded-off values of the dynamic factor. For this reason, the intermediate values of D are determined in calculations by interpolation. Let us find, for example, the value of the dynamic factor to which the point A (Fig. 9.2) at a 90% load corresponds. The inclined lines in the figure are drawn through each $0.05D$. Point A lies between the lines which correspond to the values 0.20 and 0.25 of the dynamic factor. In order to determine the value of D at a given load to an accuracy of about one hundredth, the section ab is divided into five equal portions (by the number of divisions on the scales D_0 and D_a). Point A lies approximately on the third division above the line 0.20, and therefore $D = 0.23$ in our case.

The load nomogram allows all the above problems to be solved not only for the case of full load on the motor vehicle but also for any intermediate value.

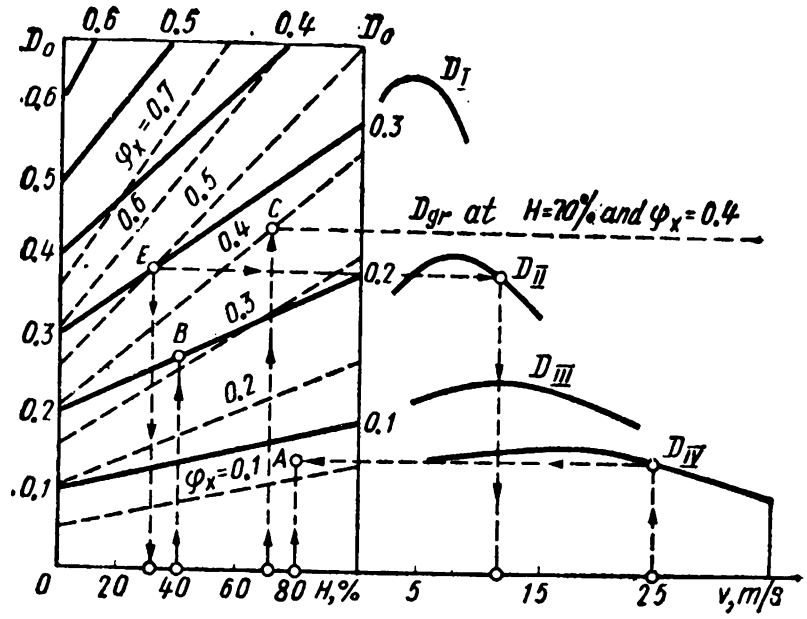
Let us determine, for example, the maximum value of the coefficient ψ when the motor vehicle carries a load equal to 40 per cent and moves at a speed of 25 m/s. With this aim in view, we draw a vertical line through point $v = 25$ m/s (Fig. 9.2) up to the intersection with the curve of the dynamic factor and a horizontal line leftwards through the point of intersection. We now draw a vertical line through the given value of load (40 per cent) until it intersects the obtained horizontal line and determine at the point of intersection the sought-for value of the dynamic factor which is equal in our case to 0.048. Hence, the maximum value of the coefficient ψ is also equal to 0.048 under the same conditions.

The problem is solved in the reverse order when determining the maximum possible motor vehicle speed. Let the load $H = 80$ per cent and the coefficient $\psi = 0.14$ be given. After drawing a vertical line through the specified value of load, we plot the value of ψ on this line and draw a horizontal line through the point obtained until it intersects the curve of the dynamic factor. The abscissa of the point of their intersection will be exactly the sought-for speed equal to 7 m/s.

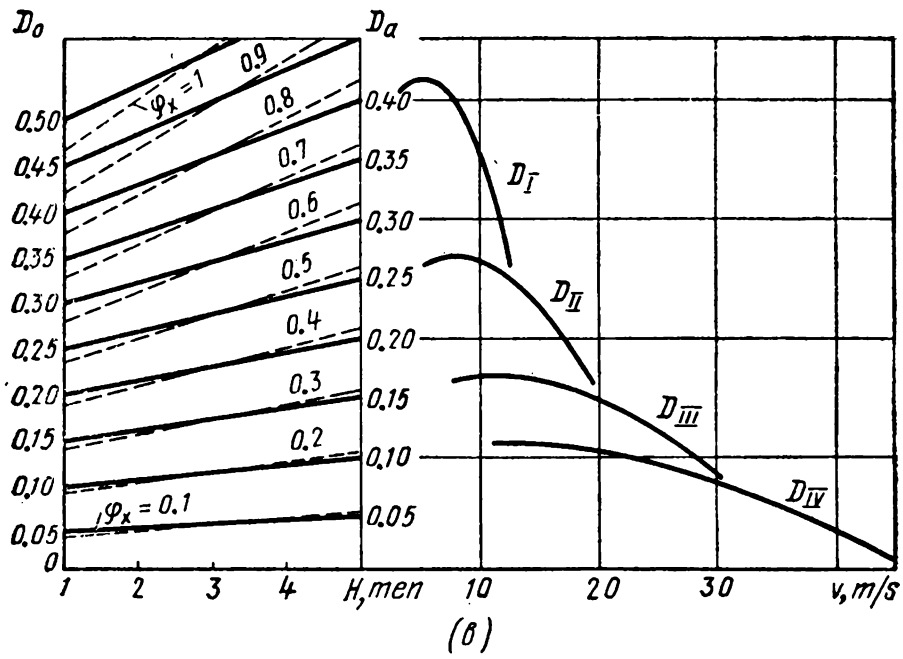
In order to find the maximum load H in a certain gear with known values of ψ and v, a vertical line is drawn through the given speed up to the intersection with the dynamic factor curve in this gear. A horizontal line is drawn from the obtained point leftwards until it intersects at point B the inclined straight line which corresponds to the specified value of ψ. The vertical line dropped from point B onto the load scale will show the value of the load H. Thus, the load is equal to 20 per cent when $v = 13$ m/s and $\psi = 0.10$. If the point of intersection of the vertical line passed through the given value of the load H and the horizontal line passing through the point on the curve of the dynamic factor is found to be between the inclined straight lines of the load nomogram, the intermediate value of the dynamic factor (or ψ) is determined by interpolation as shown above.

The horizontal line drawn through the point of intersection of the vertical line with the curve of the dynamic factor may pass either above the right-hand (upper) end or below the left-hand (lower) end of the inclined straight line which corresponds to a given value of ψ. In the first case, even with a motor vehicle under full load, the motion at a constant speed requires the throttle to be slightly shut because at full throttle $D > \psi$. In the second case uniform motion is impossible, since $D < \psi$ even at full throttle and zero load.

The skidding control graph is a graphically represented dependence of the dynamic factor as to grip on the load and makes it possible to determine the possibility of the driving wheel skidding.



(a)



(б)

Fig. 9.3. Dynamic certificate of a motor vehicle
(*a*) general view; (*b*) dynamic certificate of an examined motor vehicle

The graph is plotted as follows. First the formulas illustrated below are employed to estimate the values of the dynamic factor as to grip for a motor vehicle under full load D_{agr} and without it D_{0gr} for various values of φ_x beginning with $\varphi_x = 0.1$:

$$D_{agr} = \frac{\varphi_x G_2}{G_a} \tag{9.9}$$

$$D_{0gr} = \frac{\varphi_x G_{02}}{G_0} \tag{9.10}$$

where G_{02} is the weight taken by the driving wheels of a motor vehicle without load in N.

After this, the value of D_{agr} is plotted on the scale D_a of the load nomogram (Fig. 9.3*a*) and the value of D_{0gr} on the scale D_0, and the obtained points are connected by a straight dashed line on which the value of the coefficient φ_x (0.1) is indicated. The same method is used to determine the position of the points and mark the dash lines D_{gr} for other values of φ_x (0.2, 0.3, ..., 0.8).

With the aid of the skidding control graph, it is possible to take into account the limitations imposed on the motion of a motor vehicle by the tyre road grip of the driving wheels. For instance, the minimum value of the coefficient φ_x can be found, which is necessary for motion with pre-assigned values of load and speed, or load and coefficient ψ. In the first case the method is the same as in determining the value of D (from the known load H and speed v) but instead of D estimated by the solid inclined lines the dash lines are employed to find D_{gr}. Thus, the coefficient φ_x is equal to 0.12 at a speed of $v = 25$ m/s and a load of $H = 80$ per cent (point A in Fig. 9.3*a*). In the second case a vertical line is drawn through the known value of the load H and the value of the coefficient ψ is plotted on it, after which the inclined dash lines are used to estimate φ_x. Thus, the coefficient φ_x is equal to 0.32 at a load of $H = 40$ per cent and a coefficient of $\psi = 0.2$ (point B). It is likewise possible to evaluate the maximum coefficient ψ and speed v when the values of H and φ_x are known or determine the load H and the speed v with the known values of ψ and φ_x. Thus, the coefficient $\psi = 0.27$ with a load of $H = 70$ per cent and a coefficient of $\varphi_x = 0.4$ (point C). With this value of ψ a motor vehicle can move only with first gear and the throttle slightly closed for uniform motion. If the road resistance does not hamper the motion of the motor vehicle, it can run at any speed, even at 35 m/s. The load $H = 30$ per cent and the speed $v = 11.5$ m/s when the coefficients $\psi = 0.3$ and $\varphi_x = 0.5$ (point E).

Example. On calculating and drawing the dynamic certificate of a motor vehicle.

Let us take the value of the tractive force at various speeds from the table in the example illustrating the construction of the traction characteristics of

a car. Having determined the air resistance forces for these speeds from formula (7.36) we find the values of the dynamic factor of a motor vehicle under full load from formula (9.1). The calculation results are given in the table below.

Dynamic factor	Angular velocity of crankshaft in rad/s					
	136	220	304	388	472	556
D_{I}	0.403	0.416	0.410	0.379	0.329	0.260
D_{II}	0.260	0.267	0.261	0.240	0.206	0.156
D_{III}	0.164	0.168	0.160	0.144	0.115	0.079
D_{IV}	0.112	0.108	0.099	0.076	0.048	0.0112

Let us accept the scale a_a of the dynamic factor 4 mm: 0.01 D_a. In this case the divisions on the scale corresponding to 0.05 D_a will be arranged at a distance of 20 mm from each other. We plot the dynamic characteristic of the motor vehicle using the tabulated data (Fig. 9.3*b*).

The scale of D_0

$$a_0 = a_a \frac{G_0}{G_a} = 4 \frac{14{,}900}{17{,}900} \approx 3{,}3 \text{ mm}$$

where $G_0 = 14{,}900$ N is the own weight of the motor vehicle and the weight of the driver (750 N).

The divisions corresponding to 0.05 of the dynamic factor will be arranged on the scale D_0 at a distance of 16.5 mm. Specifying the values of the grip coefficient $\varphi_x = 0.1, 0.2, \ldots$, etc., formulas (9.9) and (9.10) can be used to find D_{agr} and D_{0gr} for a motor vehicle under full load $\left(\frac{G_2}{G_a} = \frac{9340}{17{,}900} = 0.52\right)$ and without it $\left(\frac{G_{02}}{G_0} = \frac{700}{14{,}900} = 0.47\right)$. Now we summarize the results of calculations in the table.

Designations of dynamic factor	Values of φ_x							
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8
D_{agr}	0.052	0.104	0.156	0.208	0.260	0.312	0.364	0.416
D_{0gr}	0.047	0.094	0.141	0.188	0.235	0.282	0.329	0.376

Marking the values of the dynamic factor as to grip with $\varphi_x = 0.1$ (i.e., 0.047 and 0.052) on the scales D_0 and D_a, we connect the points obtained by a dash line. Repeating the construction for other values of φ_x a skidding control graph can be obtained.

9.3. Acceleration

The time of uniform motion of a motor vehicle is usually small as compared with the total time of its operation. For instance, uniform motion in cities accounts only for 15-25 per cent of the

total time. Accelerated motion takes 30-45 per cent of the time and motion by inertia and braking, 30-40 per cent.

The dynamic characteristics of a motor vehicle in nonuniform motion are the *magnitude of acceleration* and the *distance* and *time* required for its motion within a definite range of speeds. During nonuniform running the vehicle may be accelerated and decelerated.

In order to determine the maximum acceleration from the minimum steady speed of the motor vehicle, the driver sharply

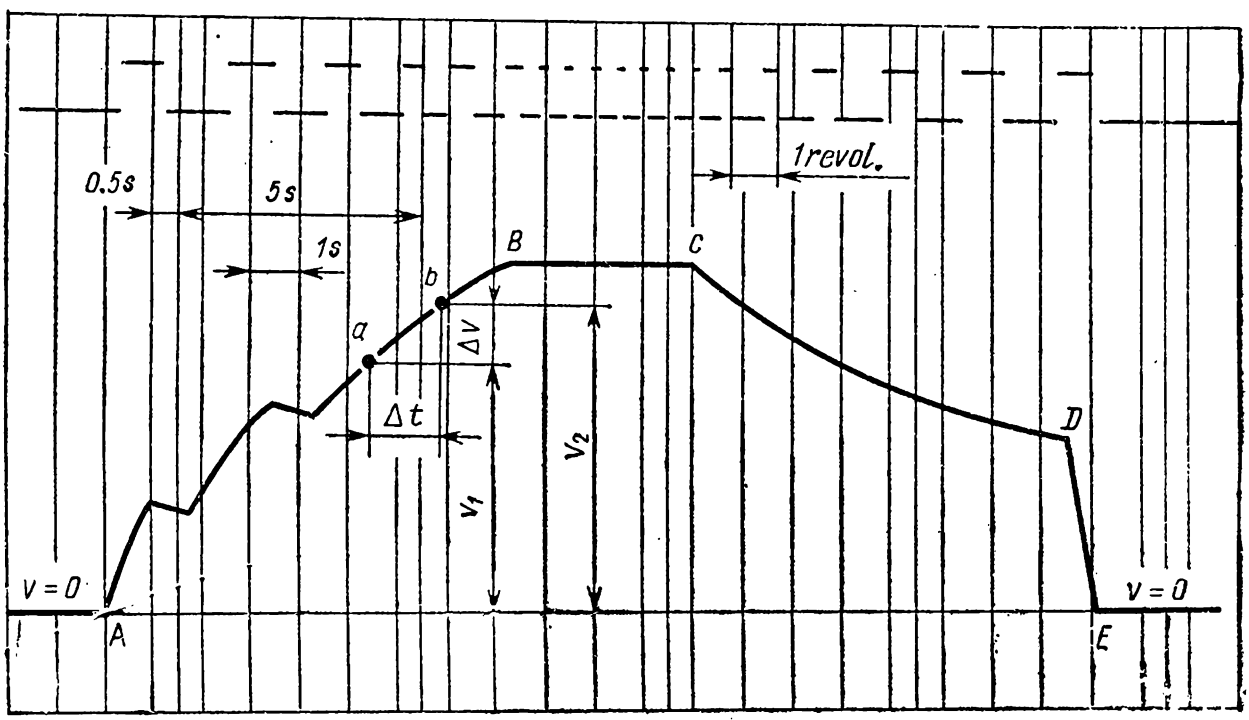


Fig. 9.4. Oscillograph tape with a record of distance, speed and time of motor vehicle motion

opens the throttle and holds it in this position until acceleration is over. The gears are changed as fast as possible in a manner which prevents the gear teeth grating in the gearbox. The change in the speed, distance and time of motion is recorded by means of recorders or a light-ray oscillograph. Distance and speed pickups are mounted on the motor vehicle wheel or on a special free-running "fifth" wheel fastened on the vehicle.

Figure 9.4 illustrates a sample of such recording made on an oscillograph tape. During recording, the tape (perforated photopaper) moves at a constant rate. The vertical lines are the marks of time placed at an interval of 1 s. The triple lines drawn after 5 s make it easier to calculate large intervals of time during prolonged tests. The horizontal dash lines make it possible to estimate the number of wheel revolutions. In the course of half a revolution of the wheel, the breaker it carries closes the loop circuit displacing the recording line downwards. During the next

half a revolution the breaker opens the circuit and the line is displaced upwards. In this manner each dash and the interval between them correspond to half a wheel revolution. Having calculated the number of wheel revolutions and knowing its radius, it is possible to find the distance covered by the motor vehicle. A tachogenerator whose voltage is directly proportional to the angular velocity of the wheel, i.e., speed of the motor vehicle, is employed as a pickup to record the curve *ABCDE* which shows the change of speed in time.

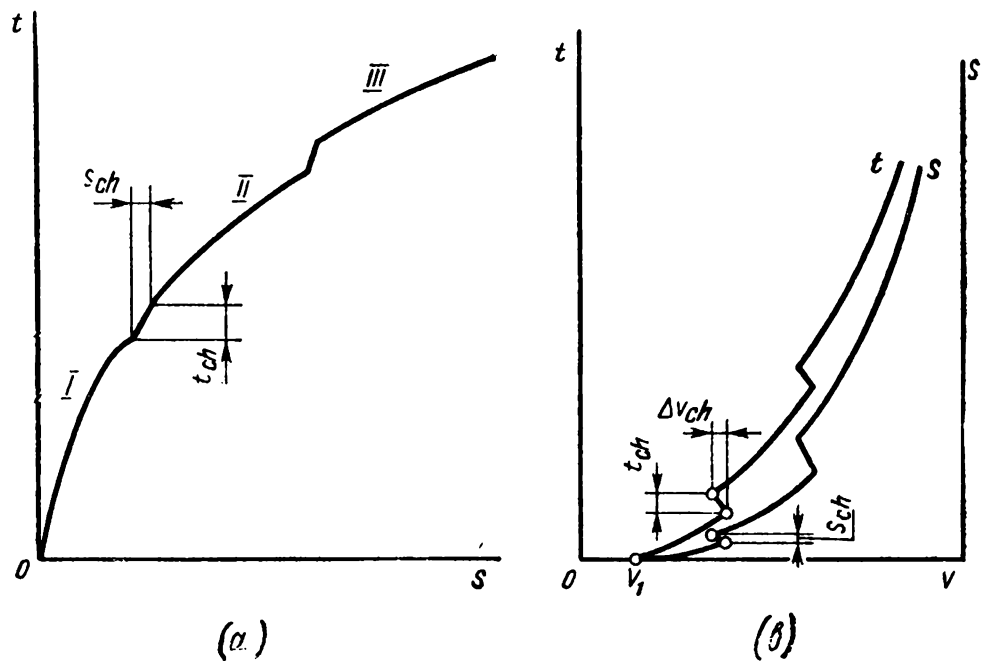


Fig. 9.5. Parameters of acceleration
(*a*) graph of acceleration rate; (*b*) acceleration distance and time graph

The section *AB* of the record corresponds to the acceleration of the motor vehicle from rest, *BC* to uniform motion, *CD* to motion under inertia and *DE* to braking. The "steps" in the section *AB* represent the reduction in the speed as the gears are changed.

When an acceleration oscillogram is processed, it is broken down into several sections with definite time intervals Δt, after which the number of wheel revolutions are assessed for each interval and the distance Δs traversed by the motor vehicle is determined. By adding the Δs values the total distance travelled during the time $t = n_s \Delta t$ (in m) can be obtained

$$s = \Delta s_1 + \Delta s_2 + \ldots + \Delta s_n \qquad (9.11)$$

where n_s is the number of intervals. The results of calculations are represented in the form of a *graph of acceleration rate* (Fig. 9.5*a*).

In order to determine the dependence of the time and distance of acceleration on the motor vehicle speed, it is usual practice

to find the speeds v_1 of the beginning and v_2 of the end of each time interval, after which the mean speed (in m/s) is estimated

$$v_m = 0.5(v_1 + v_2) \tag{9.12}$$

The results are represented in the s-v and t-v coordinates in the form of *acceleration distance and time graphs* (Fig. 9.5*b*). In comparative tests of automobiles of various types it is common practice to determine only the time of covering the first 100 metres

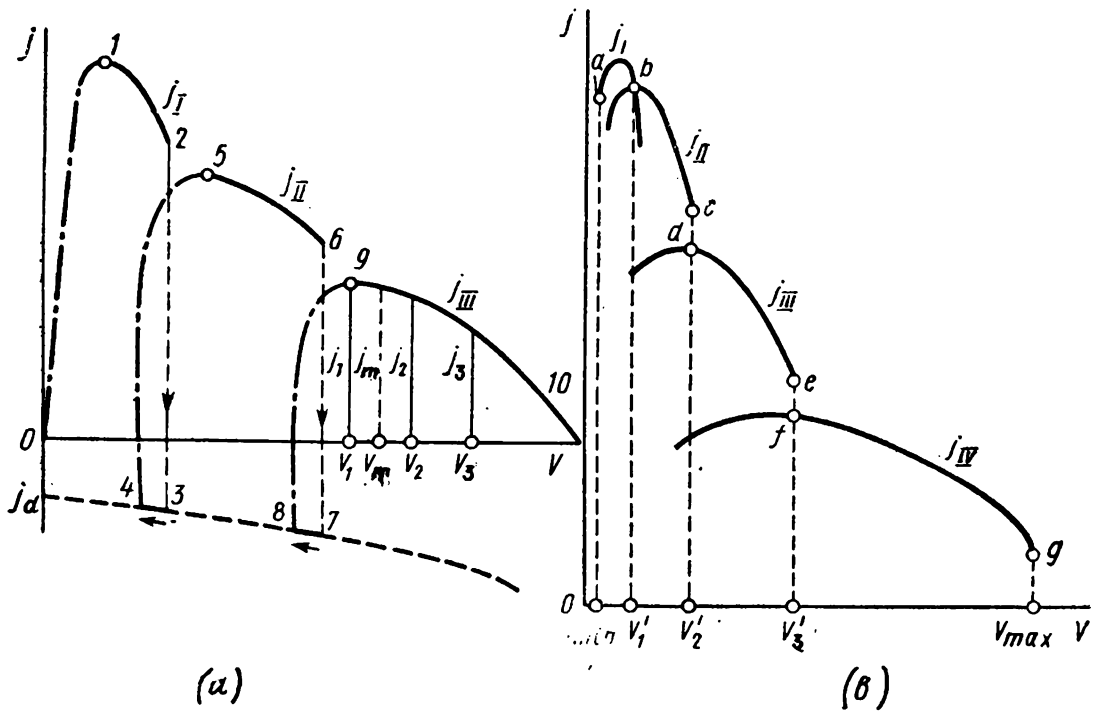


Fig. 9.6. Graphs of acceleration
(*a*) acceleration from rest; (*b*) acceleration from the minimum speed v_{min} in each gear

in acceleration from rest. This time is within 8-10 s for cars and within 10-15 s for trucks and buses.

The maximum acceleration is measured by special devices called accelerometers. If the acceleration changes during the entire acceleration period have to be estimated, the speed curve is differentiated graphically and analytically in time. Selecting a small section, for example *ab*, on this curve as shown in Fig. 9.4, the increment of speed Δv and time Δt (or distance Δs) is found in this section. Then, the mean acceleration value j_m in m/s^2 is determined for a given speed range:

$$j_m = \frac{\Delta v}{\Delta t} = \frac{\Delta v v_m}{\Delta s} \tag{9.13}$$

The dependence $j = f(v)$ can be obtained by processing the acceleration oscillogram in this manner. In plotting the graph of

accelerations, the value of j_m is referred to the mean speed within a given interval.

The graph of accelerating a motor vehicle from rest with the change of gears is illustrated in Fig. 9.6*a*. The curves above the *X*-axis show the changes in the acceleration of the motor vehicle in different gears. The curves below this axis correspond to the decelerations j_d of the automobile as it moves under its own inertia when the gears are changed.

When a motor vehicle is started the driver engages the first gear, smoothly releases the clutch pedal and simultaneously opens the throttle, thereby increasing the tractive force. In this case acceleration changes as shown by the dash-and-dot curve *0-1*. At point *1* the throttle is full open, the engine operates under full load (according to the external speed characteristic) and acceleration changes by the curve *1-2*. At the maximum speed of the motor vehicle in first gear (point *2*), the driver disengages the clutch thus disconnecting the engine from the transmission and the motor vehicle begins to decelerate (section *3-4*). Having engaged the second gear, the driver again opens the throttle and acceleration smoothly increases anew (curve *4-5*). This process is repeated also during change over to the next gears (sections *5-6*, *7-8*, etc.).

To investigate the dynamics, a motor vehicle can also be accelerated in each of the gears changing the speed from the minimum steady to the maximum possible one. A tentative graph of accelerations corresponding to such movement is illustrated in Fig. 9.6*b*. The minimum value of speed v_{min} corresponds to the minimum steady angular velocity of the crankshaft ω_{min} with full load on the engine. Within the range from 0 to v_{min} the vehicle is started from rest with the clutch slipping and the throttle opened gradually. It takes a short time to start the vehicle until the clutch is completely engaged, and it is assumed, therefore, that acceleration begins at a speed of v_{min}.

The acceleration of passenger cars at a speed of v_{max} is equal to zero, because the reserve of power in this case is also equal to zero. Trucks running at the maximum speed have a small power reserve which cannot be utilized for further acceleration, since the limiter of the maximum angular velocity of the engine crankshaft reduces the delivery of the combustible mixture.

The maximum accelerations in trucks and buses in first and second gears are about the same due to rather high gear ratios of the first gears, which sharply increase the coefficient δ_{rot}. Such vehicles must preferably be accelerated beginning with the second gear. The first gear is commonly employed to overcome high road resistances or for motion at a low speed. The approximate values of the maximum accelerations are shown in Table 9.1.

If it is impossible to estimate the dynamic characteristics experimentally, they are calculated theoretically. In this case one and the same method is used to calculate all kinds of unsteady motion (acceleration, motion under inertia, climbing uphill, etc.): first the equation of motion is utilized to find the acceleration or deceleration and then numerical integration is employed to estimate the values of time and distance.

TABLE 9.1

Maximum Accelerations of Motor Vehicles in Different Gears, m/s²

Motor vehicles	Gear	
	first	highest
Cars	2.0-2.5	0.8-1.2
Trucks	1.7-2.0	0.25-0.5
Buses	1.8-2.3	0.4-0.8

Acceleration is determined for motion on a horizontal road ($i = 0$) having a high-quality hard surface with the maximum use of engine power and without skidding of the driving wheels. The acceleration (in m/s²) can be found from formulas (8.1) and (9.2):

$$j = \frac{P_t - P_r - P_{air}}{\delta_{rot} M_a} = \frac{D - f}{\delta_{rot}} g \qquad (9.14)$$

After marking five or six values of speed on the graph of the dynamic characteristic, their corresponding D values are determined and the accelerations j are estimated from formula (9.14). The value of the coefficient f is calculated from the formula illustrated above. The graph $j = f(v)$ is plotted from the obtained values of acceleration and speed.

The time and distance of vehicle acceleration can be evaluated by a graphical and analytical method. For this purpose, the acceleration curve is divided into a number of intervals, as shown in Fig. 9.6*a* for direct drive, and it is assumed that within each speed interval the vehicle is driven with a constant acceleration of j_m. Its value can be determined from the formula:

$$j_m = 0.5 (j_1 + j)_2 \qquad (9.15)$$

where j_1 and j_2 = accelerations at the beginning and end of the speed interval, respectively, in m/s².

In order to increase the accuracy of calculation, the speed interval is taken at 0.5-1 m/s in first gear, 1-3 m/s in intermediate gears and 3-4 m/s in the highest gear.

When the speed is changed from v_1 to v_2 the mean acceleration

$$j'_m = \frac{v_2 - v_1}{\Delta t_1} = \frac{\Delta v_1}{\Delta t_1} \tag{9.16}$$

Hence, the time of acceleration within the same speed interval

$$\Delta t_1 = \frac{\Delta v_1}{j'_m} \tag{9.17}$$

The time of acceleration within the speed interval $v_2 - v_3$ in s

$$\Delta t_2 = \frac{\Delta v_2}{j''_m}$$

The total acceleration time from the minimum steady speed v_{min} to the final speed v_{max}

$$t = \Delta t_1 + \Delta t_2 + \ldots + \Delta t_n \tag{9.18}$$

The values of t determined for various speeds are used to plot the acceleration-time curve beginning from the speed v_{min} for which $t = 0$. The value of Δt_1 is plotted for a speed of v_2 and the value of $(\Delta t_1 + \Delta t_2)$, for a speed of v_3, etc.

In order to determine the maximum rate of acceleration, the calculation includes the maximum speeding-up possible at a given speed. For this reason, the time of acceleration of a truck (see Fig. 9.6*b*) is, for example, determined in the following sequence: from the speed v_{min} to the speed v'_1 by the curve j_I (section *ab*), from the speed v'_1 to the speed v'_2 by the curve j_{II} (section *bc*), from the speed v'_2 to the speed v'_3 by the curve j_{III} (section *de*) and from the speed v'_3 to the maximum speed v_{max} by the curve j_{IV} (section *fg*). The gears should be changed at the speeds v'_1, v'_2 and v'_3.

The gear change time t_{ch} depends on the skill of the driver, the design of the gearbox and the engine type. In diesel engines, the angular velocity of the crankshaft is reduced to the idle-run speed at a slower rate than in carburettor engines, because of the large inertia masses. The values of t_{ch} illustrated in Table 9.2 are typical of a highly skilled driver.

TABLE 9.2

Gear Changing Time t_{ch}, s

Gearbox	Engine	
	carburettor	diesel
Stepped without synchronizer	1.3-1.5	3-4
Stepped with synchronizer	0.2-0.5	1.0-1.5
Semi-automatic	0.05-0.1	0.5-0.8

The value of Δv_{ch} (in m/s), by which the speed of the motor vehicle falls as the gears are changed, can be found from the formula obtained by solving the equation of overrunning (see Sec. 9.5):

$$\Delta v_{ch} = 9.3 \psi t_{ch} \qquad (9.19)$$

It is tentatively assumed when calculating the acceleration distance s that the motor vehicle moves uniformly within each speed interval with a mean speed of v_m [see formula (9.12)]. The distance increment (in m) in each of the speed intervals

$$\Delta s = v_m \Delta t = \frac{v_m \Delta v}{j_m} \qquad (9.20)$$

Adding the obtained values of Δs, a summated curve s is plotted beginning it with the same speed as the curve t. The distance s_{ch} (in m) covered by the vehicle during the time the gears are changed can be determined from the formula

$$s_{ch} = v_{ch} t_{ch} \qquad (9.21)$$

where v_{ch} = mean speed of the motor vehicle during the change of gears, m/s.

When constructing the graph of acceleration rate in t-s coordinates the first to be plotted are the values of time t and distance s which correspond to acceleration from the speed v_1 to the speed v_2 and then the values of $t_1 + t_2$ and $s_1 + s_2$ for the interval v_1-v_3, etc. After this the points obtained are connected by a smooth curve.

Example. Calculate and plot the graphs of accelerations, acceleration distance and time of a car under full load on a horizontal road with an asphalt-concrete surface ($f_0 = 0.018$). The values of the coefficient f in motion at speeds higher than 16.7 m/s are estimated from formula (7.31) and the values of D are taken from the tables in the previous examples.

The coefficient accounting for the rotating masses in motion in first gear

$$\delta_{rot\ I} = 1 + \delta_1 + \delta_2 i_1^2 = 1 + 0.04 + 0.04 \times 3.51^2 = 1.53$$

In second gear $\delta_{rot\ II} \approx 1.25$, in third gear $\delta_{rot\ III} = 1.12$ and in direct drive $\delta_{rot\ IV} = 1.08$.

The acceleration

$$j = \frac{D - f}{\delta_{rot}} g$$

at a speed of 3.12 m/s

$$j_1 = \frac{(0.403 - 0.018)\, 9.81}{1.53} = 2.47 \text{ m/s}^2$$

at a speed of 5.05 m/s

$$j_2 = \frac{0.424 - 0.018}{1.53} 9.81 = 2.54 \text{ m/s}^2$$

Mean acceleration within the speed interval 3.12-5.05 m/s

$$j'_m = 0.5\,(j_1 + j_2) = 0.5\,(2.47 + 2.54) = 2.5 \text{ m/s}^2$$

The increment in speed

$$\Delta v_1 = 5.05 - 3.12 = 1.93 \text{ m/s}$$

The results of the calculations are summarized in the table, and the graph of accelerations of the motor vehicle is show in Fig. 9.7*a*.

1st gear				2nd gear			
v	D_{I}	f	j_{I}	v	D_{II}	f	j_{II}
3.12	0.403	0.018	2.47	4.85	0.260	0.018	1.90
5.05	0.416	0.018	2.54	7.85	0.267	0.018	1.96
7.00	0.410	0.018	2.51	10.80	0.261	0.018	1.91
8.90	0.379	0.018	2.31	13.80	0.240	0.018	1.74
10.85	0.329	0.018	1.99	16.80	0.206	0.021	1.45
12.80	0.260	0.018	1.55	19.80	0.156	0.0226	1.04
3rd gear				4th gear			
v	D_{III}	f	j_{III}	v	D_{IV}	f	j_{IV}
7.55	0.164	0.018	1.28	11.0	0.112	0.018	0.85
12.20	0.168	0.018	1.31	17.7	0.108	0.022	0.78
16.90	0.160	0.021	1.22	24.5	0.099	0.025	0.67
21.50	0.144	0.023	1.06	31.2	0.076	0.030	0.42
26.20	0.115	0.026	0.78	38.0	0.048	0.035	0.118
30.90	0.079	0.029	0.44	44.8	0.0112	0.042	−0.28

The negative value of acceleration shows that the maximum speed is higher than 38 m/s but lower than 44.8 m/s. The graph of accelerations is used to find $v_{\max} = 39.5$ m/s.

The time of acceleration within the first interval of speeds

$$\Delta t_1 = \frac{\Delta v_1}{j'_m} = \frac{1.93}{2.5} = 0.77 \text{ s}$$

The mean speed

$$v'_m = 0.5\,(3.12 + 5.05) \approx 4.09 \text{ m/s}$$

The distance of acceleration within the first interval of speeds $\Delta s_1 = v'_m \Delta t_1 = 4.09 \times 0.77 = 3.15$ m.

The graph of accelerations is employed to estimate the initial speeds at which the gears are changed:

$$v'_{in} = 12.4 \text{ m/s}; \quad v''_{in} = 19.2 \text{ m/s}; \quad v'''_{in} = 30.9 \text{ m/s}$$

Assuming the time of gear change to be $t_{ch} = 2$ s, let us determine the drop in speed when the first gear is changed for the second:

$$\Delta v'_{ch} = 9.3 f t_{ch} = 9.3 \times 0.018 \times 2 = 0.34 \text{ m/s}$$

When the second gear is changed for the third $\Delta v''_{ch} = 0.42$ m/s and in transfer to direct drive $\Delta v'''_{ch} = 0.54$ m/s.

The speeds at the end of change in m/s:

from first to second $v'_{end} = 12.06$,

from second to third $v''_{end} = 18.78$,

from third to direct drive $v'''_{end} = 30.36$.

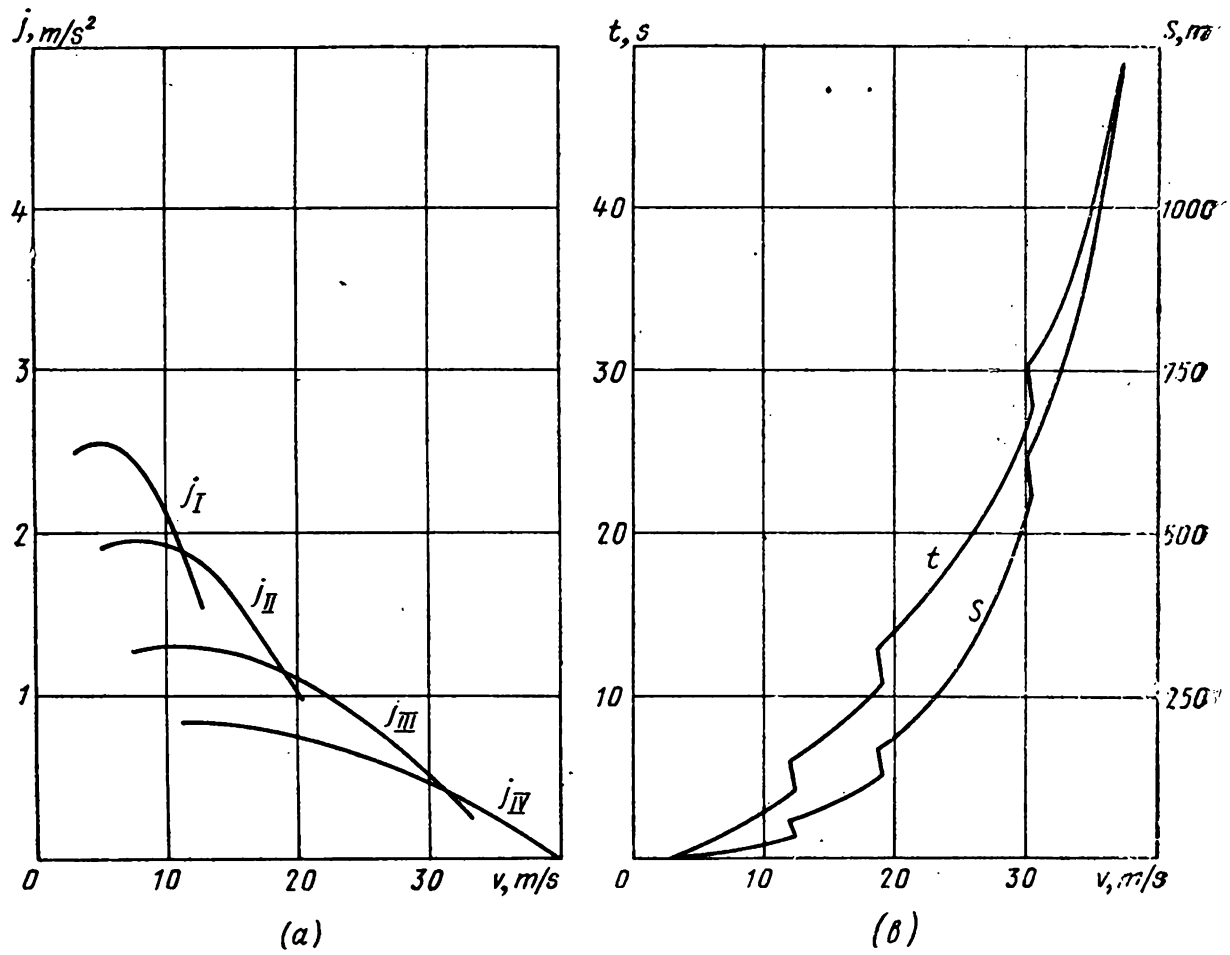


Fig. 9.7. Parameters of acceleration of an examined motor vehicle

(*a*) graph of acceleration; (*b*) acceleration distance and time graph

The mean speeds during the gear change:

$$v'_{ch} = 0.5\,(v'_{in} + v'_{end}) = 0.5\,(12.4 + 12.06) = 12.23 \text{ m/s}$$

$$v''_{ch} = 0.5\,(19.2 + 18.78) \approx 19.0 \text{ m/s}$$

$$v'''_{ch} = 0.5\,(30.9 + 30.36) = 30.63 \text{ m/s}$$

The distance covered during the gear change:

$$s'_{ch} = 12.23 \times 2 = 24.46 \text{ m}; \quad s''_{ch} = 38.0 \text{ m}; \quad s'''_{ch} = 61.26 \text{ m}$$

The results of the calculations are summarized in the table. The graphs of acceleration time and distance for the vehicle beig examined are shown in Fig. 9.7*b*.

v	j	j_m	Δv	Δt	t	v_m	Δs	s
3.12	2.47	2.50	1.93	0.77	0.00	4.09	3.15	0.00
5.05	2.54	2.52	1.95	0.77	0.77	6.03	4.65	3.15
7.00	2.51	2.41	1.90	0.79	1.54	7.95	6.3	7.8
8.90	2.31	2.15	1.95	0.91	2.33	9.88	9.0	14.1
10.85	1.99	1.93	1.55	0.80	3.24	11.60	9.3	23.1
12.40	1.88	—	—0.34	2.00	4.04	12.13	24.26	32.4
12.06	1.90	1.82	1.74	0.96	6.04	12.93	12.4	56.7
13.80	1.74	1.60	3.0	1.88	7.00	15.30	26.7	69.1
16.8	1.45	1.28	2.4	1.88	8.88	18.00	33.8	95.8
19.2	1.12	—	—0.42	2.00	10.76	19.00	38.0	126.9
18.78	1.16	1.11	2.72	2.45	12.76	20.15	49.3	164.9
21.5	1.06	0.92	4.7	5.11	15.21	23.85	122.0	214.2
26.2	0.78	0.61	4.7	7.70	20.32	28.55	220.0	336.2
30.9	0.44	—	—0.54	2.00	28.02	30.63	61.26	556.2
30.36	0.46	0.44	0.84	1.91	30.02	30.78	58.7	617.5
31.2	0.42	0.27	6.8	25.2	31.93	34.6		676.2
38.0	0.118				57.13		871.0	1547.2

9.4. Gradient Climbing

The method of estimating the maximum gradient by the dynamic characteristic, as described above, is suitable for the most arduous motoring conditions when the ascent distance is long and a uniformly moving motor vehicle overcomes it only due to the tractive force P_t. Short ascents can be climbed using the kinetic energy stored by the vehicle in acceleration. Therefore, a longer ascent distance can be covered by an accelerated vehicle than by a vehicle in uniform motion.

Let us consider the diagram shown in Fig. 9.8*a*. A motor vehicle runs at a speed of v_0 along the section AB described by the coefficient ψ_1. Acceleration begins at point B and the vehicle moves at a speed of v_1 before the ascent. In the section CE with the coefficient of resistance $\psi_2 > \psi_1$ the speed drops.

When determining the maximum ascent length it is assumed that the motor vehicle approaches the upgrade at a maximum possible speed in the section AC. The dynamic factor curve is divided into sections as shown in Fig. 9.8*b* and then formula (9.14), in which the coefficient f is substituted for the value of ψ_2, is used to estimate the deceleration and the time and distance for each speed interval. If the road resistance coefficient due to grade is equal to or less than D_{max} in a given gear, the final speed of the

vehicle is determined by the intersection point of line ψ_2 with curve D. After the speed drops to the v_{ch} value the motion will become uniform. If this coefficient on upgrade is larger than D_{max} and is equal, for example, to ψ_3 the decreasing speed will prove

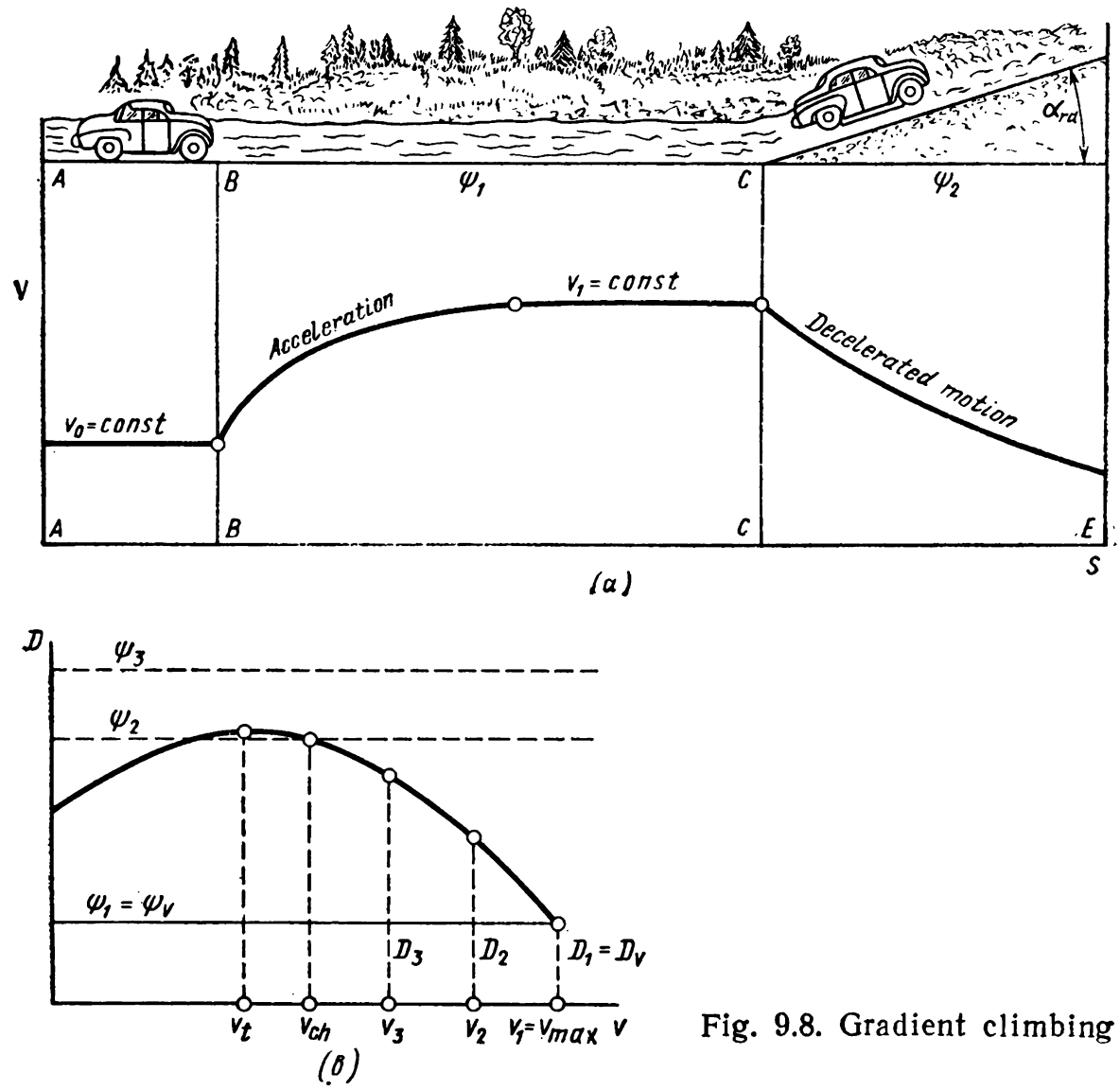


Fig. 9.8. Gradient climbing

less than the critical one and its further drop is attended by a considerable reduction in the dynamic factor.

To avoid the vehicle stopping, the lowest gear should be engaged. In this case, in the calculations, the lower speed limit is restricted by the critical speed v_t. The length l_a of ascent covered by the vehicle at the speed reduced to v_t only slightly differs from the distance at the end of which the vehicle stops.

9.5. Coasting

Coasting is frequently used in driving. This duty of motion is especially important when stoppings and subsequent accelerations regularly alternate and also in motion on roads with alternating

ascents and descents. When coasting the engine is disconnected from the transmission, the torque is not delivered to the driving wheels and the tractive force is absent. The power expended to overcome friction in the transmission is low, because all of its units operate idly.

The dynamic characteristics of a motor vehicle moving under inertia are usually determined on a horizontal road section with a hard surface. The vehicle is accelerated to a definite steady speed (for example, 14 m/s) and then on a signal from a controller the driver disengages the clutch and drive. The time, speed and distance traversed by the motor vehicle are registered by means of recorders. By way of example, Fig. 9.4 shows the coasting oscillogram.

In simplified tests, measurements are taken only of the distance travelled under inertia.

Let us write the coasting equation of a motor vehicle to calculate its dynamic characteristics:

$$\delta_{cst} M_a j = P_g + P_r + P_{air} + P_h \qquad (9.22)$$

where $P_h = \frac{M_h}{r}$ = transmission friction force applied to the driving wheels at idle run in N.

If experimental data is not available, the P_h force can be estimated (in N) from the empirical formula

$$P_h = (2 + 0.09v)\, G_a 10^{-3}$$

Equation (9.22) is solved in the P-v coordinates with the aid of the curves P_h, P_r and P_{air} adding each subsequent force upwards to the value of the previous force (Fig. 9.9*a*). The values of the force P_g, determined for several positive and negative values of gradient i, are plotted on the graph as horizontal lines, the values of P_g being plotted upwards from the X-axis at a gradient of $i < 0$ and downwards, at a gradient of $i > 0$.

The dynamic characteristics are determined from the obtained graph in the same manner as for the above force balance method. Thus, the maximum speed v_{max} is found by the abscissa of the summated curve intersection point $P_h + P_r + P_{air}$ of the resistance forces with the straight line P_g corresponding to a given gradient (for example, v_{max2} for point A). If the straight line P_g passes above the summated curve the motor vehicle accelerates and if below, decelerates.

Assuming on an average $\delta_{cst} = 1{,}05$, let us find from equation (9.22) the deceleration (negative acceleration) j_d (in m/s^2):

$$j_d = \frac{P_g + P_r + P_{air} + P_h}{\delta_{cst} G} g \approx 9.3 \frac{P_g + P_r + P_{air} + P_h}{G} \qquad (9.23)$$

Figure 9.9*b* illustrates the acceleration changes for various gradients (see the figures near the curves) when the motor vehicle coasts. The curves j, which correspond to the motion of the vehicle on upgrade, on horizontal sections and when gently descending, pass below the X-axis. Therefore, when a vehicle coasts on such sections its speed diminishes. The curves describing the vehicle motion on rather steep descents cross the X-axis at a certain

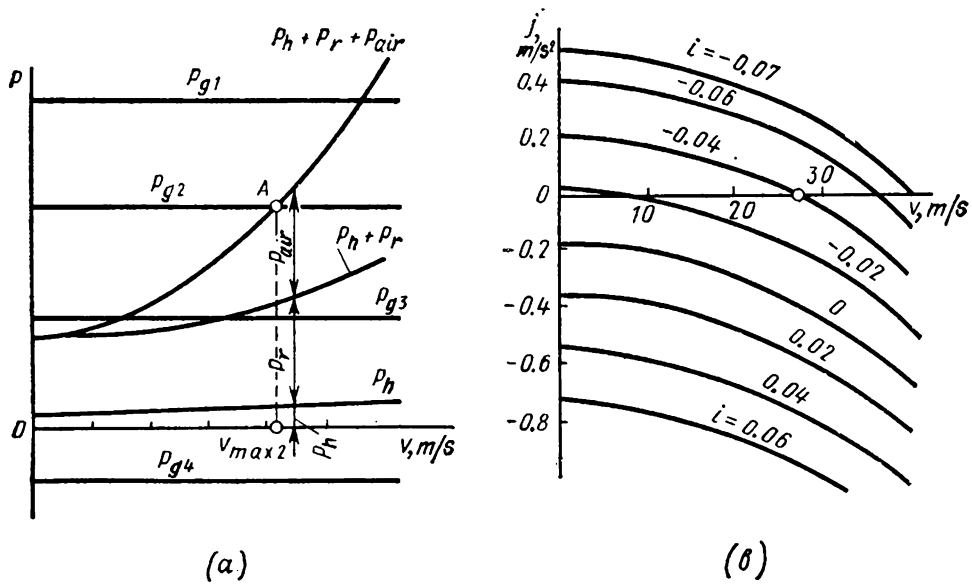


Fig. 9.9. Parameters of coasting
(*a*) force balance of a motor vehicle; (*b*) acceleration

speed. In such cases the motive forces equalize the resistance forces $P_h + P_r + P_{air}$ and the motor vehicle moves uniformly. Thus, the curve in Fig. 9.9*b* corresponding to $i = -0.04$ intersects the X-axis at a speed approximately equal to 28 m/s. If the initial speed of a motor vehicle exceeds 28 m/s, it will decelerate on such a section; if the initial speed is lower it will accelerate. However, both the decelerated and accelerated motions continue only up to the speed of 28 m/s when the vehicle begins to move uniformly. Using equations (9.17) and (9.20) it is possible to find the time and distance travelled under inertia from the known acceleration values.

In practice, the length of the running-out distance is widely employed to estimate the dynamics of a coasting motor vehicle. This distance is also a clear indication of the technical state of the vehicle chassis. The overrunning distance is longer with a better state of the chassis. The mean values of the running-out distance of Soviet-made automobiles on a horizontal section of road with a hard surface are illustrated in Table 9.3.

If a motor vehicle moves at a relatively low speed the forces P_{air} and P_h may be disregarded. Then, the deceleration of the

coasting vehicle (in m/s^2) will be

$$j_d = \frac{\Delta v}{\Delta t} = 9.3 \frac{P_{rd}}{G_a} = 9.3\psi \qquad (9.24)$$

From the latter expression formula (9.19) is obtained for evaluation of the speed reduction when the gears are changed. If the initial speed of the vehicle at the moment of gear change is equal to v_{in}, the speed at the end of change will be $v_{end} = v_{in} - \Delta v_{ch} \approx \approx v_{in} - 9.3\,\psi t_{ch}$. The mean speed of the vehicle during the time t_{ch} is

$$v_{ch} = 0.5\,(v_{in} + v_{end}) = v_{in} - 4.65\psi t_{ch}$$

In this way the distance (in m) covered by the motor vehicle as the gears are changed will be

$$s_{ch} = (v_{in} - 4.65\psi t_{ch})\, t_{ch}$$

This expression is nothing but formula (9.21) in a developed form.

TABLE 9.3

Running-out at a Speed of 14 m/s

Automobile	Running-out distance, m	Automobile	Running-out distance, m
ЗАЗ-966 "Zaporozhets"	300	ГАЗ-53А	550
"Moskvich 408" and "Moskvich 412"	450	ЗИЛ-130	600
ГАЗ-24 "Volga"	650	ЛАЗ-695	650
ЗИЛ-114	650	ПАЗ-652	550

The simplicity of experimental research into coasting has made this method highly popular to determine the road resistance coefficient ψ and the streamlining factor W_{air}.

In order to estimate the coefficient ψ the coasting is started at a speed of 3-4 m/s until the vehicle stops. At such speeds air resistance and transmission friction may be neglected and the vehicle deceleration determined from formula (9.24).

In motion up to stopping, the speed increment is numerically equal to the initial speed v_{in}. Having determined the speed v_{in} and the running-out time t from the oscillogram, formula (9.24) may be used to find the coefficient ψ:

$$\psi \approx \frac{v_{in}}{9.3t} \qquad (9.25)$$

If the section of a road is not horizontal, it is possible to determine the magnitude of the gradient. For this purpose, runs are carried out in two mutually opposite directions at the same initial speed, in which case:

for running-out uphill $\psi_1 = f + i$,

for running-out downhill $\psi_2 = f - i$.

These equations are used to find the values of f and i:

$$f = 0.5(\psi_1 + \psi_2),\ i = 0.5(\psi_1 - \psi_2)$$

If an oscillograph and other recording devices are not available the running-out time is determined by means of a stopwatch and the speed, from a calibrated speedometer.

In order to estimate the streamlining factor, running-out begins from a speed of 20-25 m/s and more. If the above assumptions are accepted the equation of a motor vehicle motion may be written as

$$\frac{G_a}{9.3} j_d = \psi G_a + W_{air} v^2 \qquad (9.26)$$

Marking a small interval of time in the high-speed region on the running-out oscillogram, the formulas illustrated above may be used to determine the increment in speed, deceleration of the vehicle and its mean speed within this interval. Then, the value of W_{air} is found from the last formula.

9.6. Overtaking

When a motor vehicle moves in a traffic flow it frequently overtakes other vehicles. This manoeuvre means occupying the adjacent lane and an increased speed of motion. Much space and time is necessary for overtaking and because of this even a slight oversight may cause a serious accident.

Overtaking comprises three phases:

movement of the vehicle to the left (driving on the right-hand side of the road is meant here and elsewhere) to occupy the lane next to the vehicle being overtaken;

motion along the left-hand lane alongside the vehicle being overtaken and ahead of it;

return to the original traffic lane in front of the overtaken vehicle.

Depending on the traffic density, overtaking may be done at a constant or with an increasing speed. Overtaking at a constant speed is typical of free noncongested traffic on country roads. In these conditions there is enough space ahead of the overtaking vehicle to allow preliminary acceleration up to a speed of v_2 greater than the speed v_1 of the vehicle being overtaken.

The time t_{ov} and the distance s_{ov} needed for safe overtaking can be found as follows.

With an oncoming vehicle moving on the opposite side of the road (Fig. 9.10).

$$s'_{ov} = s_1 + s_2 + s_3 + s_4 + s_5 \tag{9.27}$$

where s_1 and s_2 = safe intervals between overtaking and overtaken vehicles at the beginning and end of overtaking, m

s_3 = distance passed by the overtaken vehicle, m

s_4 = distance covered by the oncoming vehicle, m

s_5 = sum of the overall lengths L_1 and L_2 of the overtaking and overtaken vehicles, m

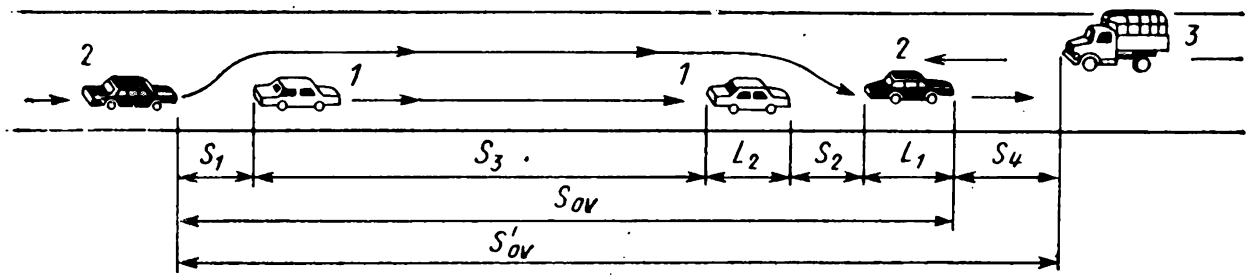


Fig. 9.10. Overtaking diagram

1—vehicle being overtaken; *2*—overtaking vehicle; *3*—oncoming vehicle

If there is no oncoming vehicle then

$$s_{ov} = s_1 + s_2 + s_3 + s_5 = v_2 t_{ov} \tag{9.28}$$

The distance (in m) traversed by the vehicle being overtaken is

$$s_3 = v_1 t_{ov} = \frac{v_1}{v_2} s_{ov} \tag{9.29}$$

From the last two equalities we obtain

$$s_{ov} = \frac{s_1 + s_2 + s_5}{v_2 - v_1} v_2 \tag{9.30}$$

The values of safety intervals depends on the speeds of the vehicles and increase as the speeds grow. The value of the first interval can be found from the empirical formula

$$s_1 = 2.5 + 0.9v_1 + 0.13v_1^2 \tag{9.31}$$

The safety interval s_2 is by 20-30 per cent less than the interval s_1, since the speed v_2 is higher than v_1 and when the overtaking vehicle returns to its lane, the distance between the vehicles rapidly increases.

To simplify the calculations, it may tentatively be assumed that the numerical values of $s_1 \approx s_2 + s_5 = 0.5\, v_2$. Then, the distance

(in m) and the time (in s) needed for the overtaking vehicle to overtake the other at a constant speed:

$$s_{ov} = \frac{v_2^2}{v_2 - v_1} \tag{9.32}$$

$$t_{ov} = \frac{s_{ov}}{v_2} = \frac{v_2}{v_2 - v_1} \tag{9.33}$$

The results from formula (9.32) are represented in Fig. 9.11 by a graph which shows the change in the overtaking distance s_{ov} depending on the speed v_2 of the overtaking vehicle (by solid lines without an oncoming vehicle and by dash lines with this vehicle). The figures near the curves indicate the speed v_1 of the vehicle being overtaken. The speed v_3 of the oncoming vehicle is taken as equal to the speed v_1.

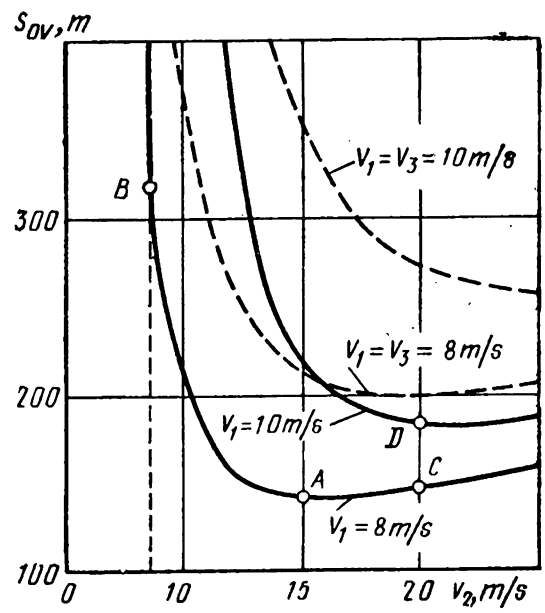


Fig. 9.11. Overtaking indices when the overtaking vehicle is accelerated

The graph shows that, when the speeds of the overtaking and overtaken vehicles approximate, the overtaking distance sharply increases. For instance, when the overtaking vehicle moves at a speed of $v_2 = 15$ m/s and the one being overtaken at $v_1 = 8$ m/s the overtaking distance $s_{ov} = 140$ m (point A). If $v_2 = 8$ m/s this distance increases to 320 m (point B).

The higher the speed v_1 of the vehicle being overtaken, the longer the overtaking distance s_{ov}. Thus, if the overtaking vehicle moves at 20 m/s and outruns a vehicle having a speed of $v_1 = 8$ m/s, the overtaking distance required will be 145 m (point C). If $v_1 = 10$ m/s the overtaking distance increases to 185 m (point D).

Overtakings at constant speed are a common feature on roads more than seven metres wide with a traffic density of about 40-50 vehicles per hour in both directions. When traffic density is high the overtakings become more complicated and dangerous. In this case a high-speed vehicle which wants to overtake a vehicle ahead, drops speed and follows it for some time at the same speed v_1 until a sufficient interval forms ahead of the vehicle to be overtaken. The driver of the high-speed vehicle then begins his outrun at an accelerated speed. If it is assumed that the overtaking vehicle during the outrun is accelerated at its maximum rate, the

minimum values of the overtaking distance and time can be determined from the graph shown in Fig. 9.12*a*.

In order to plot the graph in the t_{ov}-s_{ov} coordinates, a curve *00* is marked showing the acceleration rate of the overtaking vehicle, obtained on the basis of experimental or calculated data for full

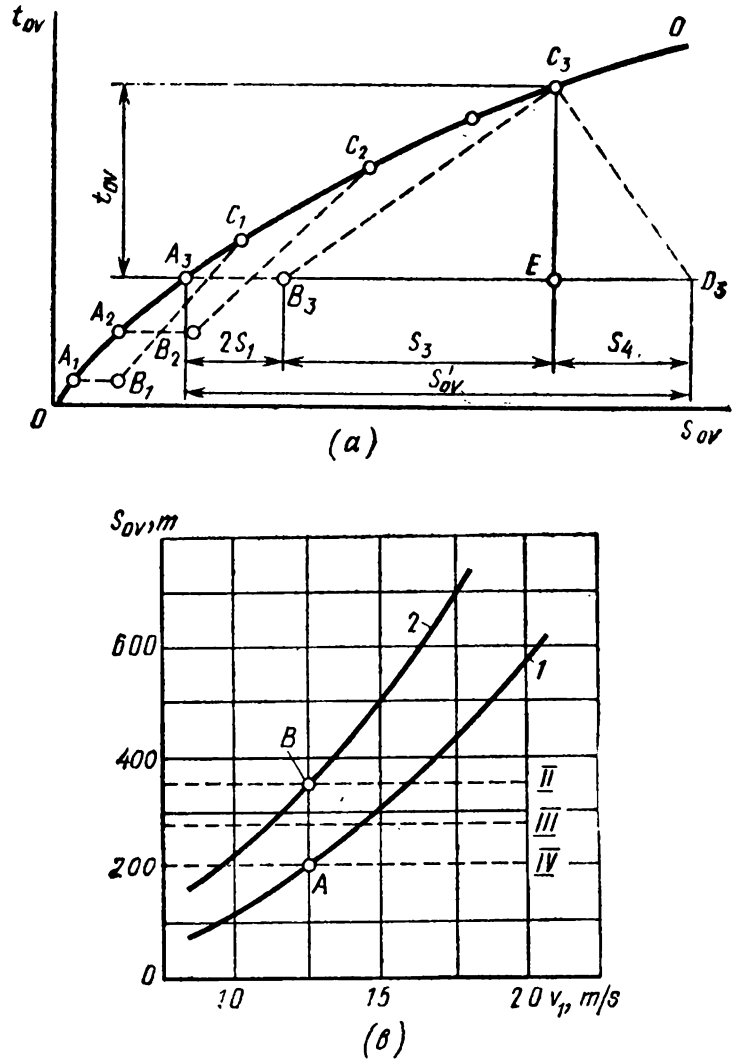


Fig. 9.12. Calculation of overtaking time and distance combined with acceleration

II-IV—road category

load on the engine. Points A_1, A_2, ..., corresponding to the speeds of 10, 20, ... m/s are marked on the curve. Sections A_1B_1, A_2B_2, ..., equal to a distance of $2s_1$ are plotted on the horizontal lines for each point. Inclined straight lines B_1C_1, B_2C_2, ..., showing uniform motion of the vehicle being overtaken at a speed of v_1 are drawn from the ends of the sections. The points C_1, C_2, ..., where the curve *00* intersects the inclined straight lines, make it possible to determine the distance and time of outrun when the overtaking vehicle runs on the right-hand side of the road. Thus, the distance s_{ov} necessary for the overtaking corres-

ponds to the section A_3E for overtaking which commences at a speed described by the point A_3.

If the road width is small and a hazard of meeting an oncoming vehicle exists, an inclined line is drawn at an angle corresponding to the speed of the oncoming vehicle (for instance, C_3D_3) up to the intersection with the horizontal line passing through the point where the overtaking begins (in our case through point A_3). The distance s'_{ov}, which should be free of traffic ahead of the overtaking vehicle, corresponds in this case to the length of section A_3D_3. The difference in the ordinates of the points A and C (for example, A_3 and C_3) illustrates the overtaking time t_{ov}.

The results of such calculations for a car ГАЗ-24 "Volga" are shown in Fig. 9.12*b*. Curve *1* corresponds to the overtaking within the limits of the right-hand side of the road and curve *2*, to the overtaking with an overtaking vehicle driving to the left side along which an oncoming vehicle moves. As can be seen on the drawing, the "Volga" car needs not less than 210 m (point A) on the right-hand side of the road for a safe overtaking when the vehicle being overtaken moves at a speed of 12.5 m/s. If the overtaking vehicle has to cross the axial line, the safe overtaking distance will be increased to 350 m (point B).

The dash lines on the same drawing show the visibility distances which should be ensured on roads of various categories. USSR standards on the road design have five road categories to suit the traffic density and speed and their importance in the entire road network. The roads of all-Union and republican importance belong to categories I-II. The roads of category III usually have regional importance and the roads of categories IV-V, local importance.

Visibility for the driver of an oncoming vehicle is only 350 m even for the roads of category II. This ensures safe overtaking only if the speed of the overtaken vehicle is 12 m/s. On the roads of category III, where a vehicle is most liable to drive for overtaking on the left-hand side, the visibility distances are so small that only vehicles moving at 8-10 m/s can safely be overtaken.

Chapter X FUEL ECONOMY OF A MOTOR VEHICLE

10.1. Fuel Economy Characteristics

Fuel is the most important material utilized by motor vehicles in large amounts. The cost of fuel amounts to 10-15 per cent of all the transportation charges. For this reason, fuel must be used to the maximum effect without unwarranted losses.

Fuel consumption of a motor vehicle depends on its design and technical state, on road and climatic conditions, skill of the driver and organization of the transportation process.

Perfection of vehicle design from the standpoint of fuel saving is estimated from the total fuel consumption Q, in litres, as referred to the length of the traversed distance s in km or the magnitude of transport operation in ton-km. The fuel consumption for a run of 100 km

$$q_{run} = 100 \frac{Q}{s} \tag{10.1}$$

The fuel consumption for unit run can be determined rather easily, but it does not take into account the transport operation undertaken by the motor vehicle, and this may result in wrong conclusions when fuel economy is evaluated. Thus, when a vehicle carries a load its fuel consumption per unit run is always greater than with the same vehicle operating in the identical conditions but without load. Therefore, it would be more correct to estimate the fuel economy of a motor vehicle by fuel consumption as referred to a unit of transport operation.

For the engine fuel economy characteristic use is made of the fuel consumption in kilograms per hour of operation G_f kg/hr and also the specific effective consumption in grams per 1 kW during one hour of operation g_eg/(kW·hr).

The hourly consumption of fuel (in kg/hr)

$$G_f = \frac{g_e N_e}{1\,000} \tag{10.2}$$

At the same time

$$G_f = 3\,600 \frac{\rho_f Q}{t} \tag{10.3}$$

where ρ_f = fuel density, kg/l

t = time corresponding to the consumption Q [formula 10.3)] or the distance s [formula (10.1)], m

From formulas (10.1) and (10.3) we obtain

$$G_f = 36\rho_f q_{run} \frac{s}{t} \qquad (10.4)$$

By solving jointly equations (10.2) and (10.4), we find the fuel consumption in litres per 100-km run

$$q_{run} = \frac{g_e N_e}{36\rho_f v} \qquad (10.5)$$

According to the USSR standard, the fuel economy road tests of motor vehicles are conducted on a horizontal straight section

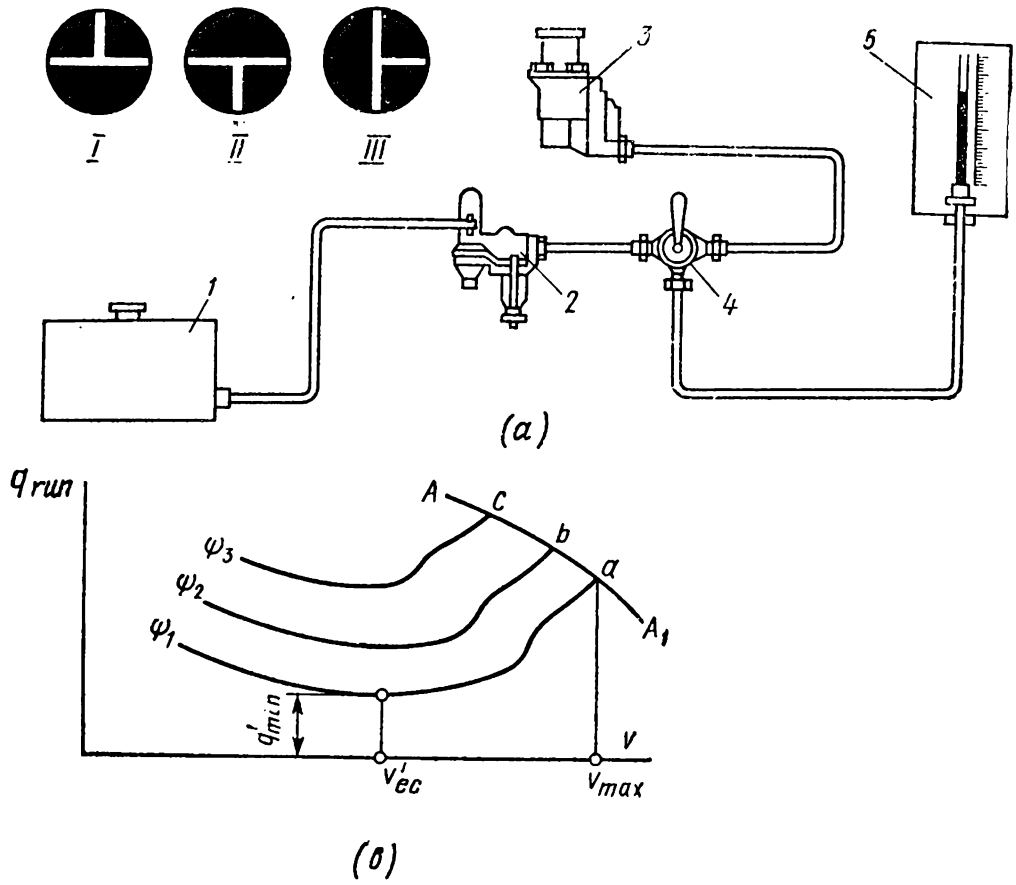


Fig. 10.1. Determining fuel economy of a motor vehicle
(*a*) diagram of measuring fuel consumption; (*b*) fuel economy characteristic of a motor vehicle; *I*, *II*, *III*—positions of the cock; *1*—fuel tank; *2*—fuel pump; *3*—carburettor; *4*—three-way cock; *5*—measuring tank

of a road one kilometre in length, which has a hard and even surface. The vehicle, carrying a full load, is accelerated outside the measured distance to a definite constant speed. As the vehicle enters the measured distance a stopwatch is started and three-way fuel cock *4* (Fig. 10.1*a*) shifted from position *I* to position *III* and fuel is delivered into carburettor *3* not from fuel tank *1* but from measuring tank *5* carrying a glass tube and a scale. The tank is connected to the fuel mains between carburet-

tor *3* and fuel pump *2*. When the vehicle leaves the measured distance the stopwatch is stopped and cock *4* returned to position *I* to cut out the measuring tank. The fuel consumption along the measured distance is determined from the difference of the levels in the tank before and after measurement. The cock is set to position *II* to restore the original level of fuel in tank *5*, after which the run is repeated at the same speed in the reverse direction.

Having determined the mean values of the specific "running" fuel consumption q_{run} and the speed v, the dependence $q_{run}=f(v)$ can be obtained with a given value of the road resistance coefficient.

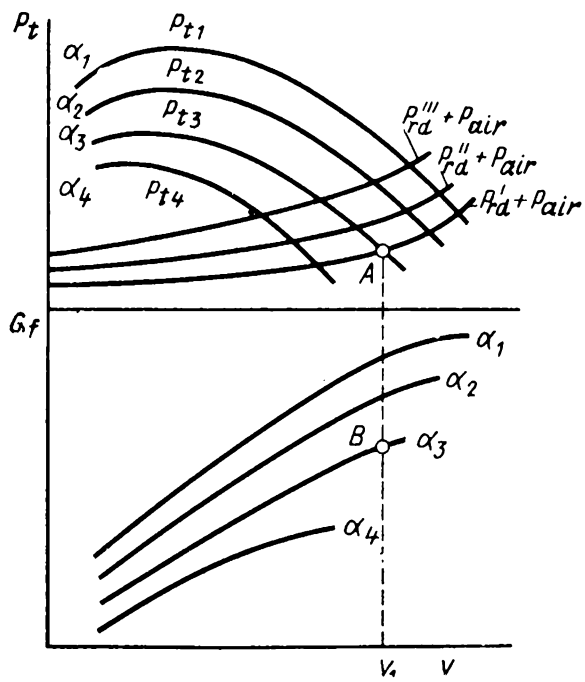


Fig. 10.2. Determining fuel consumption from the results of stand tests

After repeating the tests on the sections of the road with other values of the coefficient ψ the *fuel economy characteristic of the motor vehicle* is plotted, i.e., the graph of dependence of the specific running fuel consumption q_{run} on the vehicle speed v and the road resistance coefficient ψ (Fig. 10.1*b*). This graph describes the fuel economy of a vehicle in uniform motion and allows the fuel consumption to be estimated from the known values of q_{run} and v. Each curve on the graph has two distinctive points. One of them shows the minimum fuel consumption in motion on the road with a given value of the road resistance coefficient (for example, $q'_{\min}$ at ψ_1). The speed corresponding to this consumption (for instance, v'_{ec}) is known as *economical*. The other (finite) point on the curve shows consumption of fuel with full load on the engine, which corresponds to the maximum possible speed of motion with this road resistance coefficient (points *a*, *b*, *c*). The curve AA_1 drawn through these points represents the change of the running consumption of fuel depending on the speed v with the engine under full load. The bends in the right-hand parts of the curves are caused by mixture enrichment when the economizer is engaged.

The vehicle fuel economy is characterized by the *minimum running consumption of fuel* which corresponds to the speed v_{ec} when the vehicle under full load is tested on a horizontal section of the road with hard pavement. The control consumption of fuel in-

dicated in the technical specifications of motor vehicles practically differs but little from the minimum consumption.

When a motor vehicle is bench tested with running drums (see Fig. 7.2) the tractive force P_t and hourly consumption of fuel G_f can be obtained depending on the speed v with various positions of the throttle or the pump rack. For more convenient use, these dependences are combined on one graph as shown in Fig. 10.2. Each curve corresponds to a definite angle by which the throttle is opened (α_1, α_2, ...). Formulas (7.34) and (7.36) are employed to find the forces of road and air resistance for various values of ψ and v. This allows curves $P_{rd} + P_{air}$ to be plotted on the graph. The speeds of the motor vehicle with the throttle in a given position can be found from the points of intersection of each of these curves with the curves P_t. The curves G_f which correspond to these throttle positions are used to determine the hourly consumption of fuel. For instance, with a coefficient ψ_1 and angle α_3 (point A) the speed of the motor vehicle is v_1 and fuel consumption is estimated over the section $v_1\alpha_3$ (point B). After finding several values of G_f and v for one magnitude of the coefficient ψ formulas (10.1) and (10.5) are used to determine the running consumption of fuel and plot the curve $q_{run} = f(v)$. Repeating the construction for other values of the coefficient the graph of the fuel economy characteristic of a motor vehicle is plotted.

10.2. Fuel Consumption Equation

Let us determine the basic factors on which the running consumption of fuel q_{run} depends.

During motor vehicle motion the effective power N_e of the engine is expended to overcome the resistances to motion discussed above. It follows from the power balance equation that

$$N_e = N_t + N_{tr} = (P_{rd} + P_{air} + P_{in}) \frac{v}{1000\eta_{tr}} \qquad (10.6)$$

Comparing expressions (10.4) and (10.6) we get

$$q_{run} = \frac{g_e}{36{,}000\rho_f\eta_{tr}} (P_{rd} + P_{air} + P_{in}) \qquad (10.7)$$

The last formula shows that the fuel consumption per unit run diminshes when fuel economy of the engine estimated by the value g_e is improved. All design parameters of the engine and the features of its working process on which effective fuel consumption depends affect the fuel saving of a motor vehicle. For example, motor vehicles with diesel engines whose effective fuel consumption lies within 230-290 g/(kW·hr) are usually more eco-

nomical than vehicles with carburettor engines which consume 290-380 g/(kW·hr).

The fuel consumption grows with an increase in the resistance forces of the transmission, road and air. When a motor vehicle is accelerated, some of the fuel is spent on increasing its kinetic energy and therefore the total consumption of fuel increases.

Example. Find the running fuel consumption of a car moving uniformly at a speed of 10 m/s on a road characterized by a coefficient of $\psi = 0.081$. The specific effective fuel consumption $g_e = 350$ g/(Kw·hr) and fuel density $p_f = 0.76$ kg/m³. The transmission efficiency $\eta_{tr} = 0.9$. The values of P_{rd} and P_{air} are to be taken from the previous examples (see Secs. 7.4 and 7.5).

Fuel consumption for a run of 100 km in litres

$$q_{run} = \frac{g_e (P_{rd} + P_{air})}{36{,}000 \rho_f \eta_{tr}} = \frac{350 (1450 + 57.5)}{36{,}000 \times 0.76 \times 0.9} = 21.4$$

The practical use of equation (10.7) for estimating fuel consumption is hampered by the fact that the transmission efficiency included in this equation is not constant but depends on the conditions of motion of a motor vehicle. In conformity with formula (7.9), the less the resistance forces to motion, the smaller the tractive moment M_t and the efficiency of the transmission η_{tr}. If a motor vehicle runs at a low speed on a good road, when the forces P_{rd} and P_{air} are small, the value of η_{tr} may be much less than indicated in Table 7.1. Assuming, therefore that the transmission efficiency is constant a certain error is introduced into the value of the fuel consumption being determined. However, this has to be accepted, since introduction of a variable efficiency value into the calculations will make them much more complicated. Besides, the specific fuel consumption g_e is not an independent variable but alters with a change in the angular velocity ω_e and the degree of power utilization U of the engine as shown in Fig. 10.3. When an engine operates under a small load, the relative expense of energy to overcome the internal engine resistance increases. For this reason the curves of specific consumption in the region of small values of U sharply rise. As the values of U

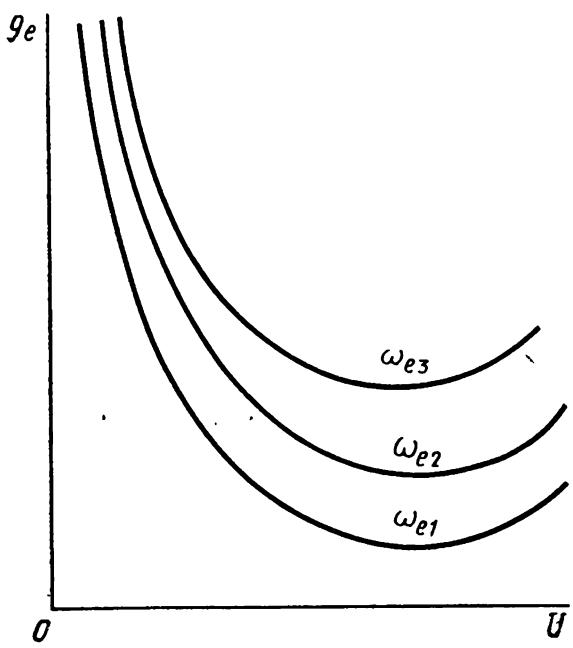


Fig. 10.3. Fuel consumption g_e versus the degree of power utilization U and the angular velocity of engine crankshaft

increase, the fuel saving improves, because the specific effective consumption drops. With high loads and almost full utilization of power, the economizer of the carburettor is engaged to enrich the combustible mixture and increase the consumption g_e. Thus, an increase or decrease in the resistance to motion tends to change the power of the engine and the speed of the motor vehicle, this tells on the value g_e and makes the fuel consumption calculation more complicated.

In order to estimate fuel consumption in definite road conditions, an experimental graph is needed (load characteristic of the engine) such as shown in Fig. 10.3.

If experimental data is not available, the motor vehicle fuel economy characteristic can be constructed using the following method.

First, the specific effective fuel consumption in g/(kW·hr) is found from the formula

$$g_e = k_U k_{\omega_e} g_N \tag{10.8}$$

where k_{ω_e}, k_U = coefficients accounting for the change in the value of g_e depending on the angular velocity and the degree of power utilization of the engine

g_N = specific effective fuel consumption at $N_{\max}$ in g/(kW·hr)

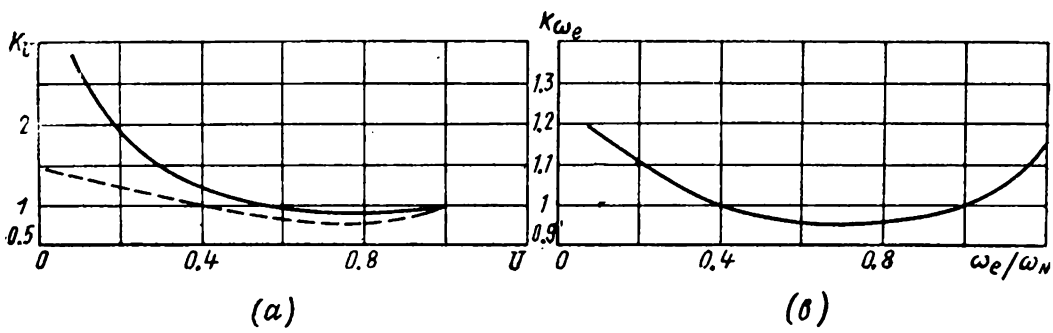


Fig. 10.4. Graphs for calculation of the fuel economy characteristic:
(*a*) graph of change in the coefficient k_U; (*b*) graph of change in the coefficient k_{ω_e}

The tentative values of the coefficients obtained by statistical processing the data are illustrated in Fig. 10.4 in which the solid line shows the change in the coefficient k_U for carburettor engines and the dash line for diesel engines. The curve k_{ω_e} refers to the engines of both types (Fig. 10.4*b*).

According to equations (10.8) and (10.7), with $P_{in} = 0$, it may be written

$$q_{run} = \frac{k_{\omega_e} k_U g_N (P_{rd} + P_{air})}{36{,}000 \rho_f \eta_{tr}} \tag{10.9}$$

To construct the fuel economy characteristic, first the graph of vehicle power balance should be constructed. After several values of the road resistance coefficient are specified, it is possible to determine the power N_{rd} and the total power $N_{rd} + N_{air}$ and then plot them on the graph (Fig. 10.5). Having divided the range of the speed change in the highest gear from v_{min} to v_{max} into several intervals, the ratio v/v_N (or ω_e/ω_N) is determined for each value of v and Fig. 10.4 is used to find the value of $k_{e\omega}$. For each value of v (or ω_e), the degree of power utilization U is estimated at a given value of the coefficient ψ from formula (8.5) and the graph of power balance, and a value of k_U is found with the aid of Fig. 10.4*a*. Upon inserting the found values of $k_{e\omega}$ and k_U into formula (10.9) we find the fuel consumption q_{run} when a motor vehicle moves on the road with a given value of the coefficient ψ. Repeating the calculations for other values, the vehicle fuel economy characteristic is plotted.

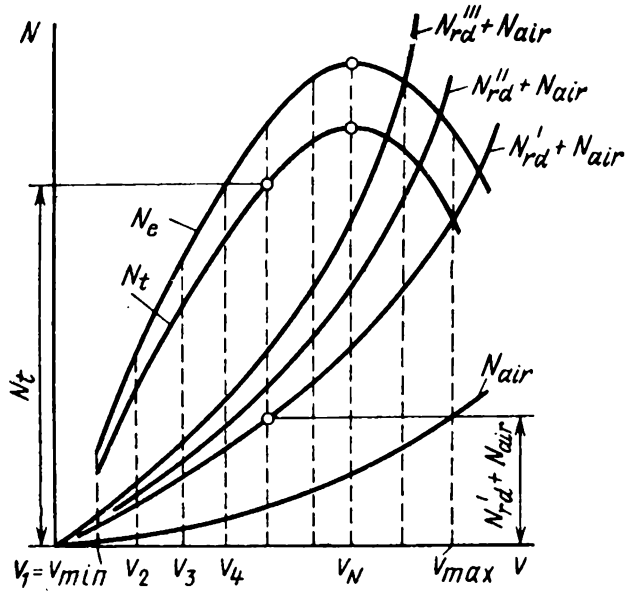


Fig. 10.5. Power balance of a motor vehicle

10.3. Effect of Operating Factors on Fuel Economy

The automobile fuel economy characteristic allows the fuel consumption to be determined for uniform motion on a road with constant gradient. It does not take into account the continuous change in the road situation and the resulting changes in the speed and load duties of the engine operation. The fuel economy characteristic corresponds to the motion of a motor vehicle at a constant speed under the action of the tractive force, whereas in actual fact the motor vehicle moves nonuniformly for most of the time, acceleration being frequently followed by coasting and sometimes by braking. The growth in speed during acceleration is attended by increases in the forces of resistance to motion and fuel consumption. However, this increases the degree of engine power utilization and reduces the specific effective fuel consumption g_e. During coasting and braking, the engine idles, a fact not accounted for in equation (10.7), which forms the basis for the construction of the fuel economy characteristic.

The motor vehicle is started from rest and moves under strenuous road conditions in low gear, with an increased value of g_e and an insufficient use of the engine power. The higher the transmission gear ratio, the higher the engine angular velocity and the less the degree of its power utilization. For this reason motor vehicle motion in low gear usually requires more fuel.

The driver selects the proper duties of motion and the moments of gear change from his own experience. Therefore, the fuel consumption usually varies within ± 10 per cent of its mean value when one and the same vehicles are driven in identical conditions by the drivers of different skill.

When driving, the combined method of acceleration and coasting is sometimes employed. The method consists in intensive acceleration in high gear to a definite speed of v_1 and subsequent coasting to a certain speed of v_2. The acceleration-coasting cycle is periodically repeated. During acceleration the engine develops a power close to the maximum one, which diminishes the specific effective consumption of fuel. The total consumption during acceleration at some mean speed is larger than in motion at the same but constant speed, because in acceleration some of the fuel is used to increase the kinetic energy of the motor vehicle. However, fuel consumption is sharply reduced in subsequent coasting and is practically equal to zero if ignition is switched off. As a result, the total consumption of fuel for identical values of the mean and constant speeds in the case of such intermittent motion may prove less than consumption at a constant speed.

Commonly, for acceleration the devices of the feed system in modern automobile engines prepare an enriched mixture. This increases fuel consumption which is not always compensated for by its decrease during coasting. Besides, the energy expended in acceleration to move the engine parts faster cannot be utilized during coasting to overcome the resistance to motion, this also impairs the fuel economy of the motor vehicle. Finally, intermittent motion is accompanied by intensive wear of parts of the engine, transmission and running gear, and fatigues the driver.

Deterioration of the vehicle technical state tends to increase fuel consumption. The fuel economy is primarily impaired as a result of wrong adjustment of the devices of the feed system and ignition and also the distributing mechanism of the engine. Thus, fuel consumption may increase by 10-15 per cent if the economizer is faulty. A deviation of the fuel level in the carburettor float chamber from the normal value also worsens fuel economy especially at low speeds. If one spark plug fails to operate the fuel consumption of a six-cylinder engine increases by 20-25 per cent; in the case of two faulty spark plugs, it increases by 50-60 per

cent. Wrong ignition timing tends to increase fuel consumption by 80 per cent.

Engine thermal conditions are likewise very important.

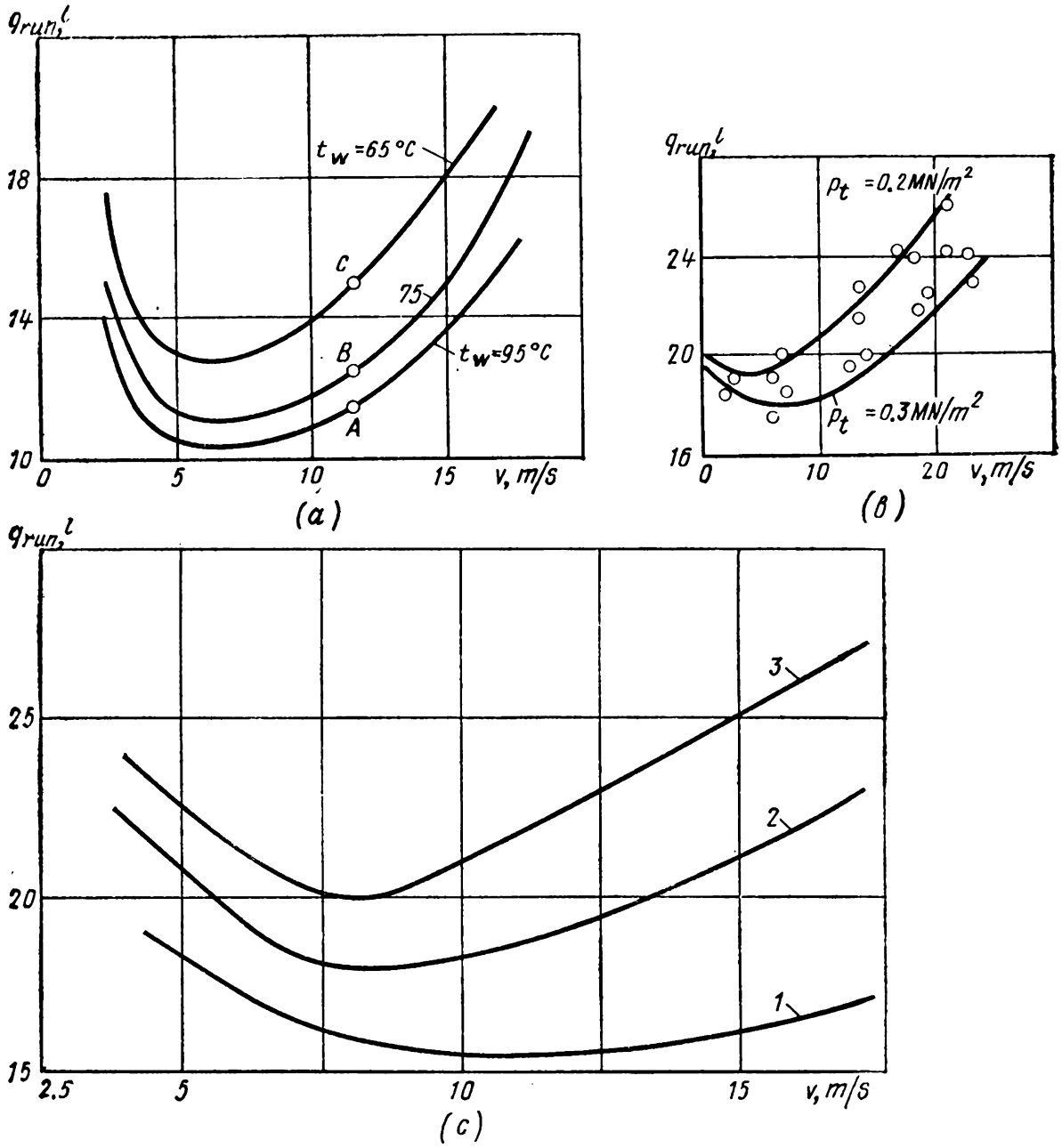


Fig. 10.6. Fuel economy characteristics of a motor vehicle
(*a*) with various water temperature in the cooling system; (*b*) with various air pressure in the tyres P_t; (*c*) with tyres of different design; *1*—broad tyres; *2*—ordinary tyres; *3*—arched tyres

Excessive cooling causes the engine to lose heat rapidly, and this diminishes the indicated efficiency. Besides, some of the fuel is admitted into the cylinder in the form of drops and fails to be burned out. All this increases fuel consumption and impairs the efficiency of a motor vehicle. Figure 10.6*a* shows fuel economy characteristics which correspond to three cooling water tempera-

tures. A drop in the water temperature from 95 to 75 °C increases fuel consumption by 6-7 per cent and a reduction to 65 °C, by almost 25 per cent (see, for example, points *A*, *B* and *C*).

The fuel economy is also affected by the technical state of the chassis units. Wrong adjustment of the main drive gears, angular contact bearings and brakes, low pressure of air in the tyres or improperly adjusted steering cause additional resistance to motion and, as a result, overconsumption of fuel.

By way of example, Fig. 10.6*b* illustrates the fuel economy characteristics of a motor vehicle with a load-carrying capacity of 2.5 tons with various air pressure in the tyres P_t. A drop in pressure from 0.30 to 0.20 MN/m^2 (at a speed of 15 m/s) increases the consumption g_{run} by almost 17 per cent.

The effect exerted by the type of tyres on fuel consumption is illustrated in Fig. 10.6*c*. As can be seen on the drawing, the replacement of ordinary tyres (curve *2*) by broad tyres (curve *1*) tends to diminish, and in the case of arched tyres (curve *3*), to increase the consumption of fuel within the entire operation range of speeds.

10.4. Fuel Consumption Rate

Fuel rating stimulates the profitability of vehicles and reduces the transportation costs.

The above formulas are convenient for the analysis of the effect of separate factors on fuel consumption, but are unsuitable for fuel rating as the formulas are rather complicated, require much work and do not take into account many factors which increase fuel consumption under operating conditions.

It has been established by theoretical and experimental research that the fuel consumption in litres per 100 km can be expressed by the equation

$$q_{run} = A_f + B_f (P_{rd} + P_{air}) \tag{10.10}$$

where A_f = fuel consumption due to internal (thermal and mechanical) energy losses in the engine, l.

B_f = increment in fuel consumption corresponding to the total force of resistance to motion equal to 1 N, l

The force of road resistance (in N)

$$P_{rd} = \psi G = \psi (G_0 + G_{load}) \tag{10.11}$$

where G_{load} is the weight of useful load in N.

Therefore,

$$q_{run} = [A_f + B_f (\psi G_0 + P_{air})] + B_f \psi G_{load} \tag{10.12}$$

Thus, fuel consumption may be regarded as a value consisting of two parts, one representing constant losses of energy in the

engine and its expenditure on the vehicle motion (expression in square brackets) and the other showing fuel consumption necessary to move the load, i.e., for the transportation work. Multiplying both sides of equation (10.12) by the run s in km and dividing them by 100 a formula can be obtained for estimating fuel consumption (in l) by the norms

$$Q = k_{f1} \frac{s}{100} + k_{f2} \frac{W_{tr}}{100} \tag{10.13}$$

where $k_{fl} = A_f + B_f (\psi G_0 + P_{air})$ = rate of consumption for the vehicle motion and internal losses in the engine

$k_{f2} = 1000 \psi B_f$ = rate of fuel consumption per unit of transportation work W_{tr}.

Fuel consumption in operating conditions is also affected by the vehicle staying at the stations of loading and unloading where the vehicle has to manoeuvre frequently moving backwards and also standing idle. Manoeuvring causes unproductive fuel expenditure while standstill cools the units (engine, transmission case, driving axle), which should be warmed up for subsequent motion. All this requires additional fuel. In order to take it into account, the equation of rating commonly includes a third term proportional to the number of runs z_e:

$$Q = k_{f1} \frac{s}{100} + k_{f2} \frac{W_{tr}}{100} + k_{f3} z_e \tag{10.14}$$

where k_{f3} = norm of additional consumption of fuel per each run in litres.

The values of k_{f1}, k_{f2} and k_{f3} depend on the type of motor vehicle and its driving conditions. Drop side trucks are used to carry load over large distances and the number of runs per 100 km is comparatively small. The fuel consumption rate k_{f3} for such vehicles is commonly included into the consumption rate k_{f1}. Dump trucks are intended for short distances, but are fully loaded and carry the load only in one direction. This makes it possible to simplify the calculations by including k_{f2} into k_{f1} and determine the total fuel consumption rate by the first and third terms of equation (10.14). The useful load of cars is small as compared with their own weight and for this reason their fuel consumption rate is referred to unit run (to 100 km).

Chapter XI BRAKING DYNAMICS OF A MOTOR VEHICLE

11.1. General on Braking Dynamics

The traction and braking properties of a motor vehicle are closely interrelated. The higher the mean speed of a vehicle, the more attention should be given to its safe motion and the better the braking properties should be. At the same time the mean speed increases with a higher acceleration rate and more effective braking.

During braking, friction transforms the kinetic energy, stored by the motor vehicle, into heat which is dissipated in the ambient air. Braking causes unrecoverable losses, because the energy absorbed by the brakes cannot be utilized for further motion. For this reason, braking inevitably increases fuel consumption. In the case of ordinary (service) braking, most of the heat is liberated in the brakes (for instance, between friction linings and drums) and in the case of full (emergency) braking, when the wheels are locked, between the tyres and the road.

A motor vehicle can be braked by several methods: without using the brake system (coasting), with the use of the brake system, and with the use of the brake system and the engine.

11.2. Experimental Estimation of the Braking Properties of a Motor Vehicle

The braking properties of a motor vehicle can be found from stand and road tests by measuring the distance, time and deceleration when braking at the maximum rate, and also the magnitude of the braking forces.

For laboratory tests use is made of special stands usually consisting of a measuring column and fluted platforms arranged at floor level. When the motor vehicle is driven onto the platforms at a speed of 3-3.5 m/s, it is sharply braked and the platforms are displaced in the direction of motion under the action of the forces of inertia. Each platform is linked by means of a lever and a hydraulic device with an indicator on the measuring column, which makes it possible to determine the braking forces acting on the wheels and evaluate the distribution of the total braking force between them.

Road tests are conducted on even, dry and clean road sections with a hard surface when the vehicle carries full load as specified by the technical specifications of the manufacturer and the USSR standard. During road tests, the braking properties of a motor vehicle can be found, for example, with the aid of a "distance-time-speed" recorder or a tachogenerator mounted on the fifth wheel, and by means of a light-beam oscillograph. The processed

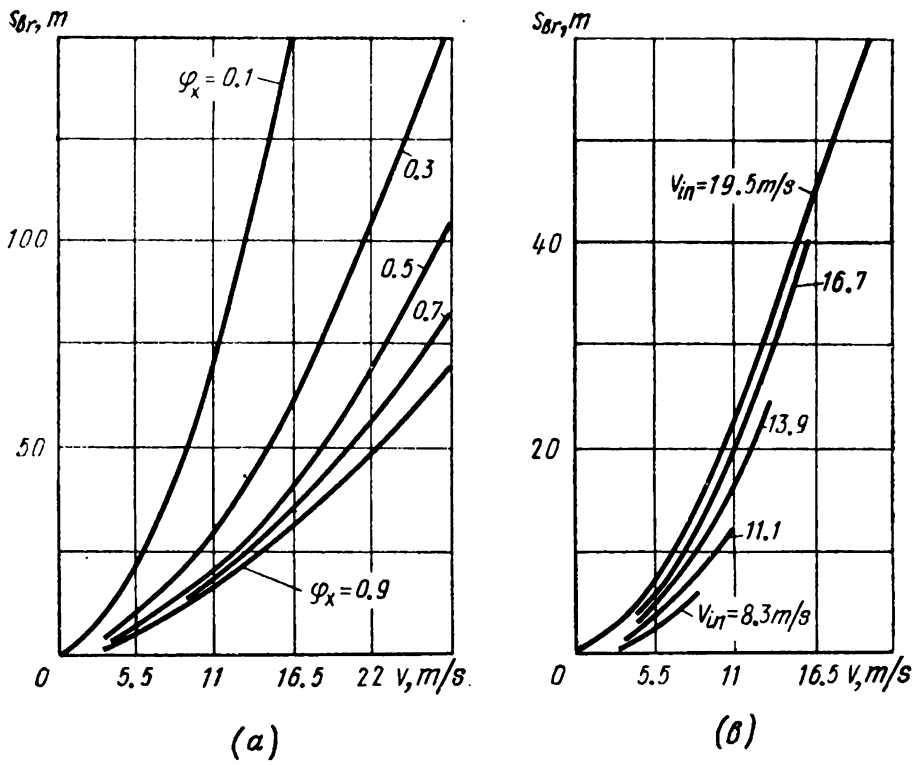


Fig. 11.1. Effect on the braking distance
(*a*) coefficient of adhesion; (*b*) initial speed of the vehicle

results of tests are employed to plot the graphs of dependence of the braking distance and time on the speed with various values of the coefficient φ_x (Fig. 11.1*a*).

From the graph it is possible to determine the braking deceleration (in m/s^2) as a function of speed from the formula

$$j_{dec} = \frac{\Delta v}{\Delta t}$$

where Δv = drop (negative increment) in speed in m/s corresponding to time Δt in seconds (in order to ensure the required accuracy the time intervals should not be larger than 0.1 s).

Figure 11.1*b* illustrates the dependence of the braking distance on speed when a motor vehicle is braked on a dry asphalted road. The graphs are plotted for various initial speeds, i.e., for speeds at which braking begins. As the initial speed grows the braking intensity is decreased due to the reduction in the coefficient of

friction between the rubbing surfaces of brakes caused by their heating.

The braking path can also be estimated with the aid of special guns which eject paint onto the road at the beginning of braking. After the motor vehicle is stopped the distance from the paint spot on the road to the gun is measured and the braking distance determined.

Decelerations in braking can be estimated by the readings of decelerometer which is a portable inertia-type device.

An idea of the change in speed and deceleration in braking is given by the diagram depicted in Fig. 11.2. The section t_r on the diagram corresponds to the *reaction time of the driver* in which he decides to apply the brake and shifts his leg from the throttle pedal to the brake pedal. The time of the driver's reaction depends on his individual abilities and skill and ranges within 0.4-1 s. In calculations, the time t_r is usually assumed as 0.8 s. The section t_{dr} shows the *brake drive operation time* from the moment the brake pedal is depressed to the appearance of deceleration. During his time, all movable elements of the brake drive are shifted. The time t_{dr} depends on the type of brake drive and its technical state and lies within 0.2-0.4 s for a hydraulic and within 0.6-0.8 s for a pneumatic drive. In auto-trains with pneumatic brakes the time t_{dr} is 1-2 s. The section t_{in} is equal (to a selected scale) to the time during which deceleration increases from zero (beginning of brake action) to its maximum value.

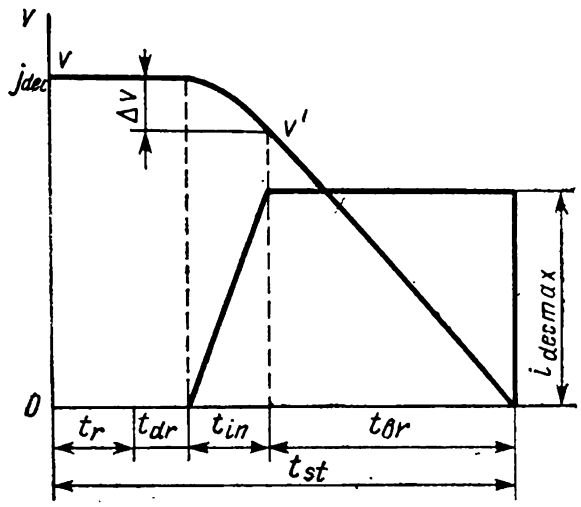


Fig. 11.2. Braking parameters

The time of deceleration increase t_{in} depends on the braking efficiency motor vehicle load, type and state of the road surface.

Below are given the values of time t_{in} (in s) for emergency braking with all wheels locked.

Cars	0.05-0.2
Trucks and buses with hydraulic brakes	0.05-0.4
Trucks with pneumatic brakes and load-carrying capacity of up to 4500 kg	0.15-1.2
Ditto with load-carrying capacity of over 4500 kg	0.2-1.5
Buses with pneumatic brakes	7.2-1.3

The smaller values specified for t_{in} are for roads with a low grip coefficient, the load on the vehicle in this case exerting prac-

tically no effect on the time t_{in}. Larger values of t_{in} correspond to motion with full load on roads with a high coefficient grip.

The section t_{br} represents the *braking time with the maximum intensity*. In the course of the time $t_r + t_{dr}$ the motor vehicle moves uniformly at a speed of v and during t_{in} with deceleration. During the time t_{br} the deceleration practically remains a constant magnitude, in view of which the speed diminishes almost by a linear law.

In order to check the effect of the action of the braking system (foot brakes), use is made of the maximum permissible braking distance and the minimum possible deceleration for vehicles without load and with a rated load (the braking intensity of cars and buses is checked for safe motion without passengers).

According to the recommendations of the UNO European Economic Commission, the braking distance should not exceed the values found from the following formulas.

For cars at an initial speed of 22.2 m/s and a pressure of 500 N applied to the brake pedal the braking distance (in m) is

$$s_{br} = \frac{v^2}{10}$$

For small and medium load-carrying capacity trucks with hydraulic brakes travelling at an initial speed of 13.8 m/s and a brake pedal pressure of 700 N

$$s_{br} = 0.54v + 0.1v^2$$

For heavy trucks and auto-trains at an initial speed of 11.5 m/s

$$s_{br} = 0.65v + 0.113v^2$$

These norms hold true for the case of braking a motor vehicle with full load on a horizontal road with a dry asphalt-concrete level surface.

11.3. Braking Force on Wheels

When braking, the elementary friction forces distributed over the friction lining surfaces build up a resultant friction moment M_{br} directed in opposition to the wheel rotation, while a braking force P_{br} appears between the wheel and the road.

The maximum value of the braking force $P_{br\,max}$ is equal to the tyre road grip force. When all the wheels of a motor vehicle are equipped with brakes (Fig. 11.3) the maximum value of the braking force (in N) is

$$P_{br\,max} = P_{br1} + P_{br2} = \varphi_x Z_1 + \varphi_x Z_2 = \varphi_x (Z_1 + Z_2)$$

But

$$Z_1 + Z_2 = G_a$$

then

$$P_{br\,max} = \varphi_x G_a \tag{11.1}$$

Since the maximum value of the coefficient φ_x lies within 0.8-0.9, the magnitude of the braking force during braking on a

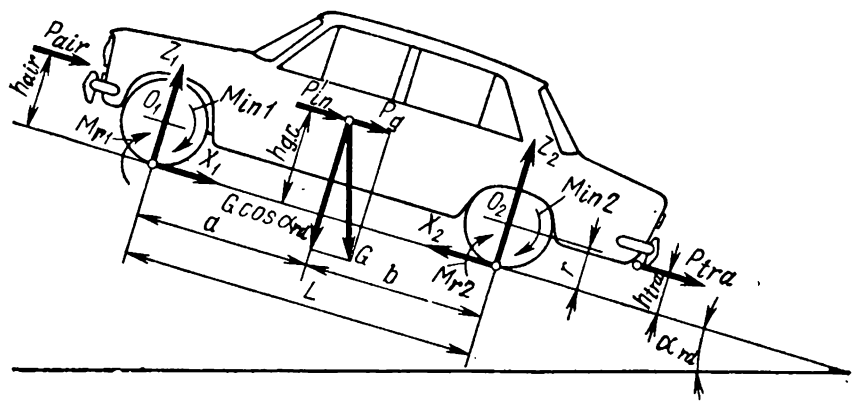


Fig. 11.3. Forces acting on a vehicle braked when travelling uphill

dry asphalted road may reach 80-90 per cent of the gravity force of the vehicle.

The braking system is normally designed so that such a braking moment M_{br} is obtained which may cause the wheels to skid when a motor vehicle with full load on a dry road with an asphalt-concrete surface is braked.

11.4. Equation of Motion of a Motor Vehicle during Braking

Let us compose a general equation of motion for a motor vehicle when it runs uphill and is braked. For this purpose, we project all the forces acting on the vehicle in braking (Fig. 11.3) onto the road surface:

$$P_{br1} + P_{br2} + P_{r1} + P_{r2} + P_g + P_{air} + P_{fe} + \\ + P_h - P_{in} = P_{br} + P_{rd} + P_{air} + P_{fe} + P_h - P_{in} = 0 \tag{11.2}$$

where

$$P_{br} = P_{br1} + P_{br2}$$

$$P_{rd} = P_{r1} + P_{r2} + P_g$$

P_{fe} = engine friction force as applied to the driving wheels, N

If the friction moment M_{fe} in the engine is known the value of the force P_{fe} (in N) can be determined from the formula

$$P_{fe} = \frac{M_{fe} i_{tr}}{\eta_{rev} r} = \frac{P'_{fe} i_{tr}}{\eta_{rev}} \tag{11.3}$$

The friction moment (in N·m) in a four-stroke engine can be determined approximately from the formula

$$M_{fe} \approx 0.8 p_m V_h i_{cyl}$$

where p_m = mean pressure of the mechanical losses estimated by formula (4.13), N/m²
V_h = working volume of the cylinder, m³
i_{cyl} = number of cylinders in the engine

Let us consider the case of braking by the brake system only when the force $P_{fe} = 0$.

Formula (11.2) will give us the deceleration value j_{dec} (in m/s²):

$$j_{dec} = \frac{P_{br} + P_{rd} + P_{air} + P_h}{\delta_{rot.\,b} M_a}$$

where M_a = mass of the automobile, kg
$\delta_{rot.\,b}$ = coefficient accounting for rotating masses during coasting and braking

Bearing in mind that the speed of a motor vehicle diminishes in braking, it may be assumed that the force $P_{air} \approx 0$. Since the force P_h is small as compared with the force P_{br}, it may be neglected especially in emergency braking. Then the accepted assumptions producing a slight error with respect to improving the braking properties of the motor vehicle permit equation (11.2) to be written in the following form:

$$P_{br} + P_{rd} - P_{in} = 0$$

Accounting for expressions (7.34), (7.45) and (11.1) we get

$$\varphi_x G_a + \psi G_a - \delta_{rot.\,b} M_a j_{dec} = 0$$

From the latter equation we obtain a transformed braking motion equation on a nonhorizontal section of road

$$\varphi_x + \psi - \frac{\delta_{rot.\,b}}{g} j_{dec} = 0 \tag{11.5}$$

11.5. Characteristics of a Motor Vehicle Braking Dynamics

For such characteristics use is made of the braking deceleration j_{dec} (in m/s²), braking time t (in s) and braking distance s_{br} (in m).

Decelariton during Braking

Solving equation (11.5) with respect to deceleration j_{dec} (in m/s²) we obtain

$$j_{dec} = \frac{\varphi_x + \psi}{\delta_{rot.\,b}} g \qquad (11.6)$$

The coefficient φ_x is usually much larger than the coefficient ψ and, therefore, the value of ψ in formula (11.6) may be neglected. In this case

$$j_{dec} = \frac{\varphi_x}{\delta_{rot.\,b}} g \qquad (11.7)$$

Taking the coefficient $\delta_{rot.\,b} \approx 1$ and the acceleration $g \approx$ $\approx$ 10 m/s² it may be assumed that in emergency braking of motor

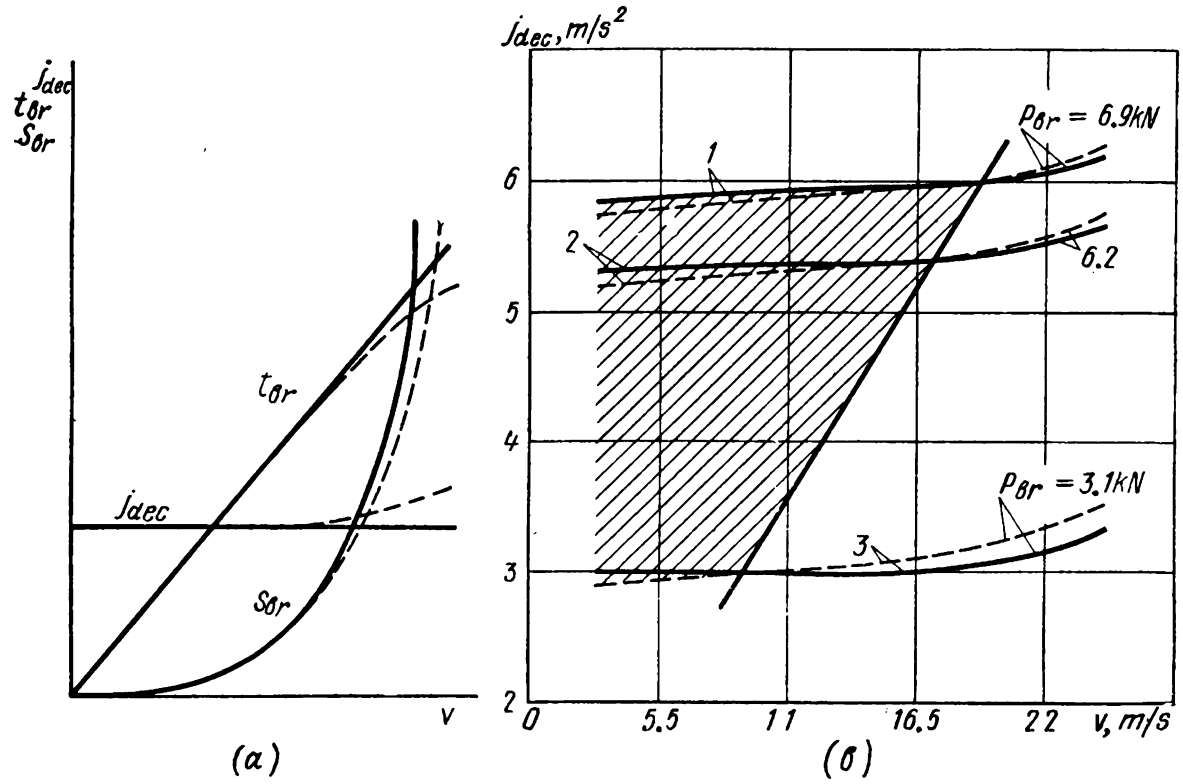


Fig. 11.4. Characteristics of braking dynamics

(*a*) deceleration, time of motion, and braking distance when braking with the brake system; (*b*) deceleration with various methods of braking; *1*, *2*, *3*—deceleration curves with various methods of braking; solid lines—braking by the brake system, dash lines—combined braking

vehicles of all models on a dry asphalt-concrete road the maximum deceleration will be within 8-9 m/s². Emergency braking, when the deceleration magnitudes are very high, causes unpleasant sensations in passengers and increased wear of brakes and tyres. For this reason, decelerations do not, as a rule, exceed 1.5-2.5 m/s² in service braking.

If the coefficient φ_x does not change during braking, deceleration remains almost constant, i.e., it does not depend on the speed of the motor vehicle (Fig. 11.4*a*).

Braking time. The braking time can be determined by graphical and analytical integration of the deceleration curve $j_{dec} = f(v)$, plotted from equation (11.4). After finding decelerations for several values of speed from the graph $j_{dec} = f(v)$ the mean deceleration is estimated within each speed interval. Then, formula (9.17) is used to determine the time increment Δt within each interval and, summing up the obtained values of Δt, to estimate the braking time. The graph of the relation $t_{br} = f(v)$ is illustrated in Fig. 11.4*a*.

Let us revert to Fig. 11.2*a* to find the time t_{st} needed to stop the motor vehicle from the moment danger arises, i.e. the so-called stopping time. During the time t_{in} deceleration changes according to a law close to a linear one. It may therefore be assumed that during the time t_{in} the motor vehicle decelerates at approximately 0.5 $j_{dec.\,max}$. In this case the speed increment (in m/s) will be

$$\Delta v = \frac{j_{dec.\,max} t_{in}}{2} \tag{11.8}$$

Hence, the speed (in m/s) at the beginning of emergency braking with deceleration $j_{dec.\,max}$

$$v' = v - \Delta v = v - \frac{j_{dec.\,max} t_{in}}{2} \tag{11.9}$$

If the further motion of the vehicle is equally retarded with a deceleration of $j_{dec.\,max}$ and stops at the end of braking the speed diminishes in the course of the time t_{br} according to a linear law from v' to zero. Then,

$$v' = j_{dec.\,max} t_{br} \tag{11.10}$$

Solving this equation with respect to the time t_{br} (in s) and replacing the speed v' by its value from expression (11.9) we get

$$t_{br} = \frac{v'}{j_{dec.\,max}} = \frac{v}{j_{dec.\,max}} - \frac{t_{in}}{2} \tag{11.11}$$

Thus, the stopping time (in s)

$$t_{st} = t_r + t_{dr} + t_{in} + t_{br} = t_r + t_{dr} + \frac{t_{in}}{2} + \\ + \frac{v}{j_{dec.\,max}} = t_{tot} + \frac{v}{j_{dec.\,max}} \tag{11.12}$$

where

$$t_{tot} = t_r + t_{dr} + \frac{t_{in}}{2}$$

If the values of the braking forces on all wheels of the motor vehicle simultaneously reach the maximum grip values the deceleration can be found from formula (11.7). In this case, taking

the coefficient $\delta_{rot.\,b} \approx 1$ with an accuracy sufficient for practical calculations we obtain

$$t_{st} \approx t_{tot} + \frac{v}{\varphi_x g} \tag{11.13}$$

Braking distance. The braking distance can be determined by graphical and analytical integration of the curve $t_{br} = f(v)$ assuming that the motion of the motor vehicle is equally retarded in each speed interval. Breaking the curve $t_{br} = f(v)$ into several intervals, formula (9.20) is employed to determine the distance increment Δs within each speed interval in the same manner as in estimating the acceleration distance. Summing up the obtained values of the distance increment Δs, the total braking distance s_{br} is evaluated. A tentative graph of the relation $s_{br} = f(v)$ with account of the forces P_r, P_{air} and P_h is shown in Fig. 11.4 by the dash line and disregarding these forces, by the solid line.

The distance necessary to stop the motor vehicle from the moment danger arises, i.e., the stopping distance, can be found assuming the nature of the deceleration change shown in Fig. 11.2.

As before, let us assume that when the speed decreases from v to v' the vehicle moves with a constant deceleration equal to $0.5\, j_{dec.\,max}$. Then, in conformity with formula (9.20), the distance covered by the vehicle during this period will be

$$s_{in} = \frac{v_m \Delta v}{j_m} = \frac{(v + v')(v - v')}{2} \frac{1}{0.5 j_{dec.\,max}} = \frac{v^2 - (v')^2}{j_{dec.\,max}} \tag{11.14}$$

During emergency braking the deceleration is equal to $j_{dec.\,max}$ and the braking distance traversed by the vehicle when the speed drops from v' to zero (in m) will be

$$s_{br} = \frac{(v')^2}{2 j_{dec.\,max}} \tag{11.15}$$

The found value of s_{br} is the distance traversed by the vehicle during the time t_{br} with the maximum braking intensity. The stopping distance s_{st} (in m) is larger than the distance s_{br} because it also includes the distance traversed by the vehicle during the driver's reaction time, operation of the brake system and the increase to deceleration rate

$$s_{st} = v(t_r + t_{dr}) + s_{in} + s_{br} = v(t_r + t_{dr}) + \\ + \frac{v^2}{j_{dec.\,max}} - \frac{(v')^2}{2 j_{dec.\,max}} \tag{11.16}$$

Inserting the speed v' from formula (11.9) into formula (11.16) and neglecting the very small value of the product containing t_{in}^2 we get

$$s_{st} = v t_{tot} + \frac{v^2}{2 j_{dec.\,max}} \tag{11.17}$$

If the braking forces on all the vehicle wheels simultaneously reach the grip forces, then, with account of equation (11.4) and with $P_{fe} = P_h = 0$, the braking distance will be

$$s_{br} = \frac{\delta_{rot.\,b} M_a (v')^2}{2[(\varphi_x + f) G \cos\alpha + G \sin\alpha + P_{air}]} \approx \frac{\delta_{rot.\,b} (v')^2}{2\varphi_x g} \approx \frac{v^2}{2\varphi_x g} \quad (11.18)$$

The braking distance is directly proportional to the square of the vehicle speed at the moment braking begins. For this reason, with higher initial speeds the braking distance increases very rapidly (see Fig. 11.4*a*). If the value of s_{br} is known the stopping distance can be found from the formula

$$s_{st} = v t_{tot} + \frac{v^2}{2\varphi_x g} \quad (11.19)$$

The values of t_{st} and s_{st} can be determined from formulas (11.12) and (11.17) if the vehicle maximum deceleration estimated experimentally is known. If the maximum deceleration is unknown the values of t_{st} and s_{st} can be found from formulas (11.13) and (11.19) if φ_x has been established.

11.6. Braking Force Distribution Between the Wheels

When a motor vehicle is braked the inertia force P_{in} acting on the arm $h_{g.\,c}$ (see Fig. 11.3) redistributes the loads on the front and rear wheels: the load on the front wheels increases and on the rear ones diminishes. Therefore, the values of the normal reactions Z_1 and Z_2 acting respectively on the front and rear wheels during braking differ appreciably from the loads G_1 and G_2 which they commonly take in the static state. As noted above, these changes are estimated by the coefficients m_{r1} and m_{r2} of the change in the reactions (see Chapter VII). If a vehicle is braked on a horizontal road the coefficients m_{r1} and m_{r2} are estimated from the formulas:

$$m_{r1} = 1 + \frac{\varphi_x h_{g.\,c}}{b}; \quad m_{r2} = 1 - \frac{\varphi_x h_{g.\,c}}{a}$$

The maximum values of the reaction change coefficients m_{r1} and m_{r2} during braking are within the limits 1.5-2 and 0.5-0.7, respectively.

Having determined the coefficients m_{r1} and m_{r2}, it is possible to find the values of the normal reactions in Neutons:

$$Z_1 = m_{r1} G_1; \quad Z_2 = m_{r2} G_2$$

The maximum braking intensity can be ensured only if the adhesion weight of the vehicle is utilized to the full. Since the ratio

between the braking forces P_{br1} and P_{br2} acting on the front and rear wheels does not change, the total force P_{br} may attain its maximum value without the wheels skidding only on a road having a grip coefficient equal, for example, to φ_0. On other roads it is impossible to utilize the adhesion weight to the full advantage without locking the front or rear wheels.

The coefficient β_0 of the braking force distribution on the front and rear wheels can be found from the formulas

$$\beta_0 = \frac{P_{br1}}{P_{br}} = 1 - \frac{P_{br2}}{P_{br}}$$

The optimum distribution of the braking force will be such when the front and rear wheels can be locked simultaneously. In this case the coefficient

$$\beta_0 = \frac{b + \varphi_0 h_{g.c}}{L} \tag{11.20}$$

Most of the brake systems uniformly distribute the braking force between the wheels on a road with a definite value of the coefficient φ_0. On other roads either the front or rear wheels are the first to skid during braking. However, a recent development is a brake system with a static and dynamic adjustment of the braking force distribution.

Since the distribution of the total braking force between the wheels does not correspond to the normal reactions that change during braking, the actual minimum braking distance is by 20-40 per cent higher than the theoretical one. In order to approximate the calculation results to experimental data, the formulas contain a coefficient K_{eff} which accounts for the degree of use of the full theoretically possible effect of the brake system action. The effective braking coefficient K_{eff} is equal on average to 1.2 for cars and to 1.4-1.6 for trucks and buses. In this case the design formulas will take the form:

$$t_{st} = t_{tot} + K_{eff} \frac{v}{\varphi_x g} \tag{11.21}$$

$$s_{br} = \frac{K_{eff} v^2}{2\varphi_x g} \tag{11.22}$$

$$s_{st} = v t_{tot} + K_{eff} \frac{v^2}{2\varphi_x g} \tag{11.23}$$

Example. Find the braking distance and time and also the deceleration of a ГАЗ-24 "Volga" car braked with a shut-off engine from a speed of 20 m/s to stopping-down on a dry horizontal road with a concrete surface ($\varphi_x = 0.6$) if the coefficient $K_{eff} = 1.2$, $t_r = 0.8$ s, $t_{dr} = 0.2$ s and $t_{in} = 0.5$ s.

The braking distance from a speed of 20 m/s to stopping-down

$$s_{br} = \frac{K_{eff} v^2}{2\varphi_x g} = \frac{1.2 \times 20^2}{2 \times 0.6 \times 9.81} = 40.7 \text{ m}$$

Stopping time

$$t_{st} = t_r + t_{dr} + \frac{t_{in}}{2} + \frac{K_{eff} v}{\varphi_x g} = 0.8 + 0.2 + \frac{0.5}{2} + \frac{1.2 \times 20}{0.6 \times 9.81} = 5.3 \text{ s}$$

Deceleration of the vehicle

$$j_{dec} = \frac{\varphi_x g}{K_{eff}} = \frac{0.6 \times 9.81}{1.2} = 4.9 \text{ m/s}^2$$

11.7. Braking Methods

Combined Method of Braking a Motor Vehicle by the Brake System and the Engine

With this method, the braking moment on the wheels is produced by the brakes and the engine simultaneously. Since in this case the throttle control pedal is released before the brake pedal is depressed, the angular velocity of the engine crankshaft would be reduced to the idling angular velocity. However, in actual fact, the driving wheels rotate the crankshaft through the transmission. This generates an additional force P_{fe} resisting motion, which is proportional to the engine friction force P'_{fe} and decelerates the vehicle. The flywheel inertia acts contrary to the engine braking effect. Sometimes the flywheel opposition proves to be greater than the engine braking action, and the braking intensity is slightly decreased.

Let us discuss the cases when combined braking is profitable.

Let us denote the deceleration in combined braking by $j_{dec.\ com.\ br}$ and the deceleration with the engine shut off by $j_{dec.\ br}$. Then, equation (11.2) may be written in the following form:

$$j_{dec.\ com.\ br} = \frac{1}{\delta_{rot}} \frac{P_{br} + P_{rd} + P_{air} + P_{fe} + P_{tr}}{M_a} \qquad (11.24)$$

where P_{tr} = force of friction in the transmission when torque is transmitted, N

δ_{rot} = coefficient which accounts for the rotating masses determined from formula (7.44) in our case

During braking with a shut-off engine, the coefficient $\delta_{rot} = \delta_{rot.\ b}$ and is therefore estimated from formula (7.47).

In this case

$$j_{dec.\ br} = \frac{1}{\delta_{rot.\ b}} \frac{P_{br} + P_{rd} + P_{air} + P_h}{M_a} \qquad (11.25)$$

Combined braking, by the brakes and engine, is more intensive than when using the brake system only if

$$j_{dec.\ com.\ br} > j_{dec.\ br}, \text{ i. e.,}$$

$$\frac{P_{br} + P_{rd} + P_{air} + P_{fe} + P_{tr}}{\delta_{rot}} > \frac{P_{br} + P_{rd} + P_{air} + P_h}{\delta_{rot.\ b}} \qquad (11.26)$$

Thus, the expediency of braking by the brake system alone or combined braking with the brakes and engine depends on the values of the forces resistant to motion P_{rd}, P_{air}, P_{tr}, P_h, P_{fe} and also on the braking force P_{br} and the coefficients δ_{rot} and $\delta_{rot.\,b}$.

Figure 11.4*b* shows graphical solution of equality (11.2*b*) for a small-displacement car with the following values of the coefficients: $\delta_{rot} = 1.08$; $\delta_{rot.\,b} = 1.04$; $\varphi_x = 0.6$. The values of the force P_{fe} are taken from experimental data.

Curves *1*, *2* and *3* in Fig. 11.4*b* correspond to different values of the braking force P_{br}. The figure shows a shaded area within which combined braking is irrational in the given conditions because the decelerations $j_{dec.\,com.\,br}$ (dash lines) prove to be less than the decelerations $j_{dec.\,br}$ (solid lines). However, combined braking on roads with a low grip coefficient increases the transverse stability of the motor vehicle with respect to skidding. In this case the stability is improved due to a more uniform distribution of the braking forces by the differential gear between the driving wheels of the motor vehicle.

Braking a Motor Vehicle with the Brake System Intermittently Shut Off

This method of braking assures maximum intensity. At the same time this method is recommended only for highly skilled drivers, since experience and much attention are required to prevent skidding. When rolling, the braked wheel accepts a greater braking force than when it skids because the grip coefficient sharply diminishes in the latter case. As the tyre skids on the road, the particles of the tread material in contact with the road become heated and softened. If the driver presses the brake pedal several times, each time slightly releasing it, some new less heated elements of the tyre come in contact with the road, and the maximum value of the gripping force is preserved. At the moment the wheel begins to skid, the driver removes some pressure from the brake pedal, the wheel rolls on the road, and the part of the tyre tread which did not previously participate in braking comes in contact with the road.

11.8. New Features in the Design of Brake Systems

During recent years the traffic safety problem has become very important due to the growth in the motor vehicle fleets and speeds of motion. Safety depends on traffic organization, the state and type of road surface, skill of the driver and construction of the vehicle.

Antilocking devices which recently came in use are the most radical means to prevent wheel skidding. Today, theoretical and experimental research is conducted to study the effect of the antilocking devices on the braking process.

The reliability of the hydraulic brake system can be improved with the aid of the brake drive divider which automatically cuts off the faulty section of the drive.

Traffic safety is increased, the speed during descents limited and the wear of the main brake system diminished by using a *decelerator* (additional brake system) mounted on buses and trucks of medium and high load-carrying capacity. By their design all decelerators can be classified as *mechanical*, *pneumatic*, *hydraulic*, and *electrodynamic*.

Mechanical decelerators are similar to the usual type of brake systems but have larger dimensions and more efficient cooling. The most popular decelerator of this type is made in the form of a disk transmission brake placed in a housing whose inner space is filled with water and connected with the cooling system.

The principle of action of pneumatic decelerators is based on the use of engine friction to build up a braking moment. Forced rotation of the crankshaft generates a braking moment due to friction in the engine and pumping losses during the admission and exhaust strokes, i.e., the resistance to the flow of mixture and gases through pipes and valves. The engine braking moment can be increased if the exhaust pipe is shut by a valve, and the engine is shifted to compressor duty after stopping fuel supply. This moment will be still higher if a higher resistance is built up at the intake by closing the air valve at the air input into the engine intake system. As a result, the engine braking moment will increase on average by 70-80 per cent as compared with the ordinary engine braking moment. The transmission increases the engine braking moment so much that this moment on the driving wheels is enough to maintain constant speed of the motor vehicle on downhill.

The hydraulic decelerators usually take the form of a hydraulic clutch designed as an independent unit installed behind the transmission case in vehicles with a mechanical transmission, or a hydraulic clutch built into a hydromechanical transmission case. This type of decelerator creates a braking moment on the universal-joint shaft. The rotor (pump) of the decelerator is rigidly linked with the driven shaft of the transmission, and the stator (turbine) is secured to the vehicle frame. During braking, the kinetic energy stored by the vehicle is transformed in the hydraulic clutch into heat energy and warms up the fluid. The hydraulic clutch is usually cooled by water admitted from the engine cooling system. The magnitude of the braking moment of the hydraulic

clutch depends on the angular velocity of the rotor and the amount of the incoming fluid. At a low speed hydraulic decelerators are little effective; besides, they are large in mass.

The electrodynamic decelerators are induction-type clutches operated by inducing eddy currents in the armature. The armatures of the decelerators may be bell-shaped or disk-shaped. By way of example, a decelerator with a disk-shaped armature is described below. The stationary housing of such a decelerator accommodates electric magnets. The armature consists of two metallic disks rigidly coupled with the driven shaft of the transmission and having ribs on the external surface for better cooling. The braking moment depends on the gap (not less than 1.5-2 mm) between the armature and the poles of the electric magnets.

If electric current flows along the winding of the electric magnet during braking the armature rotating in the magnetic field generates eddy currents producing their own magnetic field. The interaction of the magnetic fields builds up a braking moment applied to the armature. The magnitude of this moment depends on the speed of the motor vehicle. Decelerators of this type do not require adjustment or servicing and are more effective than hydraulic decelerators. But they are expensive and heavy and when braking consume much electric power (0.5-1.2 kW). As to popularity, hydrodynamic decelerators are second after pneumatic decelerators.

Braking Force Regulator

A brake system should ensure the maximum possible braking effect without losing the vehicle stability or controllability on various roads and with various loads. This can only be done if all the wheels are braked simultaneously up to their limit with the aid of an automatic *antilocking device*. The probability of locking the rear wheels and skidding in braking can be reduced considerably if the braking forces acting on the rear wheels are lessened while increasing the braking intensity or reducing the load. For this purpose the double-circuit hydraulic brake system is provided with a device which automatically restricts the pressure exerted on the rear brakes in the braking process. This increases the braking effect.

11.9. Technical Examination of Traffic Accidents

Technical examination of traffic accidents is carried out by a team of experts. A motor vehicle being driven along a street or road may cause an accident and injure pedestrians and damage

other vehicles, structures and loads or inflict any other detriment.

The aim of a technical examination is to give a scientifically substantiated characteristic of all phases of an accident, establish its causes and elucidate the behaviour of participants. The expert works to find the technical causes of accidents which may be due to defects of the motor vehicle in question, determines the time of their appearance, the speeds of motion, braking and stopping distances, etc.

The initial examination data is found from inspection of the place of accident, the motor vehicles, and interrogation of the participants and witnesses.

Knocking down pedestrians is one of the most frequent road accidents. When investigating such cases the first thing assessed is the speed of the motor vehicle at the moment before the brakes were applied and whether it was possible to employ emergency braking to prevent the accident.

Likewise, close attention is given to the surroundings and all circumstances are recorded which help to establish the conditions and nature of movement of the people involved in the accident. The place of accident with all kinds of traces and objects found (glass fragments, traces of braking, parts of the motor vehicle, etc.) is depicted schematically to scale.

If the trace of skidding is detected on the pavement the initial speed of the vehicle can be found as follows. On the site of accident the length s_{sk} left by the skidding tyres on the pavement is measured. An experiment is conducted to estimate the value of the grip coefficient φ_x or this value can be determined from tables after establishing the type and state of the road surface from an examination of the accident zone. Having established the K_{eff} coefficient formula (11.7) is used to find the value of the maximum deceleration, which could be attained by the vehicle upon emergency braking. It may be assumed with a slight error that the skidding trace forms on the road surface (beginning of full slip of the tyres) precisely at the moment of maximum deceleration. In this case the speed of the vehicle (in m/s) at the moment preceding the braking can be determined by solving the quadratic equation:

$$\frac{1}{2j_{dec.\,max}} v^2 - 0.5 t_{in} v - s_{sk} = 0$$

obtained by the combined solution of formulas (11.9) and (11.15) and neglecting the terms containing t_{in}^2:

$$v = 0.5 j_{dec.\,max} t_{in} + \sqrt{2 s_{sk} j_{dec.\,max}} \qquad (11.27)$$

By way of example let us consider a road accident when a pedestrian was knocked down (Fig. 11.5). The motor vehicle was moving at a distance of s_p from the foot-path at the edge of which stood the pedestrian P. When the vehicle and the pedestrian were the s distance apart the pedestrian began to cross the road at right angles to the vehicle motion. In Fig. 11.5 the positions of the vehicle and the pedestrian at this particular moment are marked by the figures *I*. The driver braked his vehicle but could not avoid hitting the pedestrian (position *III*) and moved in a braked

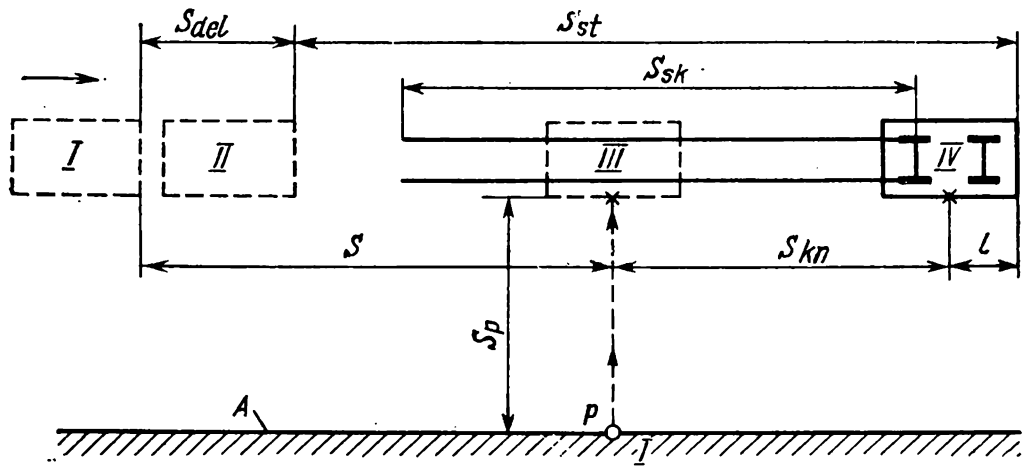


Fig. 11.5. Position of the pedestrian and motor vehicle in a road accident
A—curb

state further and stopped (position *IV*). The cross shows the point of collision. The measurements at the place of accident have shown the skidding trace length s_{sk} and the distance s_p walked by the pedestrian on the road.

The experts assumed that during the entire occasion the pedestrian could be seen by the driver who had time to prevent the accident.

In the course of examination the walking time of the pedestrian t_p is compared with the vehicle motion time t_{kn} from the moment the hazard of knocking down appeared. If the time $t_{kn} > t_p$ the driver reacted timely and could not avoid the accident by braking. If the time $t_{kn} < t_p$ he failed to use all means to prevent it.

A tentative sequence of calculations during an examination is illustrated below. The scheme of the accident is employed to determine the motion of the motor vehicle s_{kn} after knocking down in a braked state and find its speed (in m/s) at the moment of impact:

$$v_{kn} = \sqrt{2 s_{kn} j_{dec.\,max}} \qquad (11.28)$$

Then the speed of the vehicle before braking is estimated from formula (11.27), the length of the stopping path s_{st}, from formula

(11.19) and the time (in s) of the vehicle motion before it knocks down the pedestrian:

$$t_{kn} = t_r + t_{dr} + 0.5t_{in} + \frac{v - v_{kn}}{j_{dec.max}} \tag{11.29}$$

In expert calculations, the speed of a pedestrian is usually taken as constant and the path width of his movement is disregarded. With these assumptions the time (in s) of crossing the road by the pedestrian will be

$$t_p = \frac{s_p}{v_p} \tag{11.30}$$

The condition for timely braking of a vehicle may be written thus:

$$t_{kn} \geqslant t_p \tag{11.31}$$

If this condition is not fulfilled, i.e., the time $t_{kn} \leqslant t_p$, the driver was too slow in his actions and the vehicle had time to move from position *I* to position *II* (Fig. 11.5).

To find whether the driver could avoid the accident, if he had not permitted this delay but had acted timely, the time of delay (in s) is determined

$$t_{del} = t_p - t_{kn} \tag{11.32}$$

During this time the motor vehicle moving with an initial speed of v would have run a distance of

$$s_{del} = vt_{del} \tag{11.33}$$

According to Fig. 11.5, the distance s between the front of the motor vehicle and the pedestrian, as the latter begins to cross the road, is

$$s = s_{st} + s_{del} - s_{kn} - l \tag{11.34}$$

where l is the distance from the collision point to the front of the motor vehicle.

If the calculations show that the distance s is larger than the stopping path s_{st} the driver could stop his vehicle by emergency braking before the walk line of the pedestrian. If the distance $s < s_{st}$ the driver could not help knocking down the pedestrian after braking because the pedestrian had begun to move at too close a distance to the vehicle or moved too quickly.

The technical examination is also used to study the cases when motor vehicles hit stationary objects, topple over and collide. The examination results and other information are used at the inquiry as court evidence.

Chapter XII TRAILER TRAINS

12.1. Dynamic Characteristics of a Trailer Train

The trailer train is a transport facility composed of a truck-tractor and trailers or a semitrailer.

Truck-tractors may be ordinary drop-side trucks or special vehicles (towing or carrier-type) employing units of mass produced trucks or wheel tractors.

Towing vehicles operating with trailers have a coupling link, and articulated vehicles (tractor semitrailers) have a coupling support for the front of the semitrailer to rest upon. For easier manoeuvring in restricted places the carrying portion of such a vehicle is reduced as much as possible as compared with a mass produced vehicle. A towing truck is sometimes loaded with balast to increase the adhesion weight and avoid the driving wheel skidding.

The coupling link of a towing vehicle (hook, clip) is designed with a damping spring mounted on the underframe cross members. The coupling support of a carrier-type truck-tractor consists of a bearing plate which takes the vertical load and a coupling mechanism transmitting the tractive force to the semitrailer.

The running gear of the trailer towed by a truck-tractor takes the form of a frame with axles and wheels, or runners (in sledge trains) carrying a body or a swivel. Trailers with swivels employed to transport long loads (timber, pipes, rails, etc.) are called *pole trailers.*

Semitrailers differ from trailers in that the weight of the front portion of a semitrailer is taken by the coupling support of the truck-tractor, and the weight of the rear portion, directly by the rear axle or the double-axle bogie of the semitrailer.

The bodies of trailers and semitrailers, in the same manner as the bodies for motor vehicles, are made universal in the form of a load platform with drop sides, or specialized (dump-truck, tank or van). Lately, passenger trailers are also in use.

The use of trailer trains increases the load-carrying capacity of the motor fleet, especially on roads having hard surfaces without steep and long ascents.

Ordinarily, trucks moving at an average speed possess a large reserve of power utilized to overcome the increased road resistance or for intensive acceleration. For example, a power of about 30 kW

is needed for a fully loaded truck ЗИЛ-130 to move at a speed of 11-14 m/s on a good level road, whereas the engine can develop a power of 70 kW and more. For this reason the use of trailers in the case of low resistance to motion and rare stops allows the engine power to be utilized to the best advantage without any noticeable drop in the speed of a trailer train, as compared with a single motor vehicle. However, if the weight of a trailer train is excessively high, this may tend to drastically reduce its speed and efficiency, and the frequent use of the step-down gears, increase the fuel consumption.

Motion of a trailer train depends on the increasing forces of resistance to rolling and of the air. Imperfect swivelling gear and gaps in the coupling links bring the path of motion of a trailer or semitrailer out of alignment with the path of motion of the truck-tractor. A high speed tends to generate angular oscillations of trailers in a horizontal plane, and cause longitudinal jerks and shocks when the speed of the truck-tractor is changed. As a result, the force of resistance to rolling of a trailer train does not increase in proportion to its weight because the coefficient of resistance to rolling of a trailer train also increases with a growth in weight

$$f_{t\text{-}t} = \frac{f_a G_a + f_{tra} G_{tra}}{G_a + G_{tra}} \tag{12.1}$$

where f_a and f_{tra} = coefficients of resistance to rolling of the truck-tractor and trailers, respectively, and
G_a and G_{tra} = their weights in N

The coefficient f_{t-t} appreciably increases as the driving wheels tend to slip on the road when the tractive force grows.

Trailers also increase the air resistance because of an intensive turbulence in the air streams and larger surface of friction. If the distance between the truck-tractor and the trailer is comparatively small the streamlining coefficient of a trailer train k_{t-t} is larger by 8-10 per cent than in a single truck. If the distance between them is 50-80 cm the coefficient k_{t-t} grows by 15-17 per cent, and if the distance is 120-180 cm, by 30-35 per cent. It may be assumed on average that each trailer with a pole of modern design increases the streamlining coefficient of a trailer train by 25 per cent.

The movement of a trailer train is analyzed by means of its dynamic characteristic with a load nomogram (dynamic certificate of a trailer train). As distinct from a single motor vehicle, the load nomogram for a trailer train is plotted against its weight (Fig. 12.1). The weight of the truck-tractor with full load is taken as 100 per cent. In order to determine the load on a trailer train,

the weight of the loaded trailers is conditionally added to the weight of the truck-tractor.

The dynamic factor of a trailer train can be found from the formula

$$D_{t\text{-}t} = \frac{P_t - P_{air}}{G_{t\text{-}t}} = \psi + \frac{\delta_{t\text{-}t}}{g} j \qquad (12.2)$$

where δ_{t-t} is the coefficient accounting for the revolving masses of the trailer train.

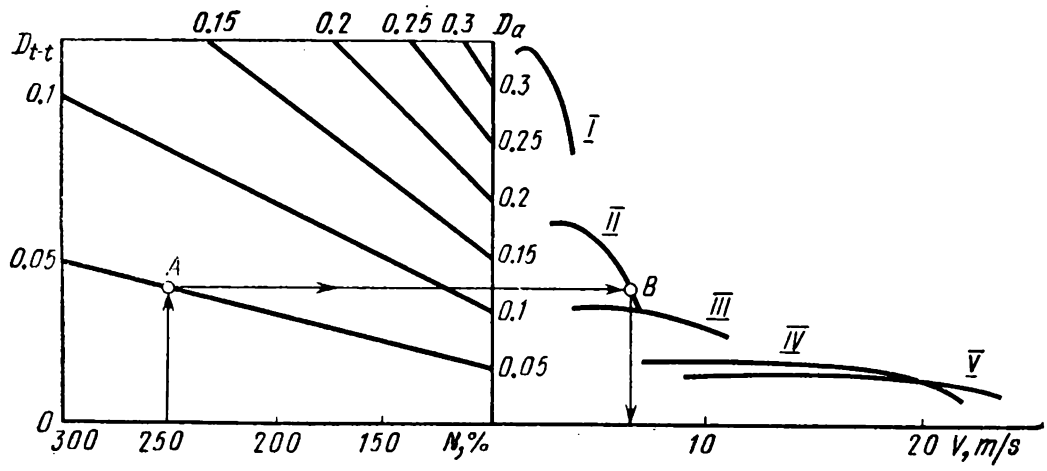


Fig. 12.1. Dynamic certificate of a trailer train
I-V—gears

Accepting for simplicity that the force $P_{air} = 0$ for the case of uniform motion we obtain:

$$G_{t\text{-}t} = \frac{P_t}{G_{t\text{-}t}} = \psi \qquad (12.3)$$

The motion of a trailer train is calculated with the aid of its dynamic certificate in the same way as that of a single motor vehicle. Thus, a trailer train exceeding the weight of a single motor vehicle by 2.5 times can move on the road described by the coefficient $\psi = 0.05$ (point *A* in Fig. 12.1) only in second gear (point *B*) with a speed of about 6 m/s.

The conducted tests and gained experience testify to the fact that the speed of a trailer train consisting of one truck and one trailer is less on a good road by about 10 per cent as compared with a single truck. For all that, the capacity of the trailer train in this case is 40-50 per cent higher that of the single truck.

Let us consider the case when a trailer train is braked. By way of example, we discuss braking of a truck-tractor with a trailer which has no brakes, the deceleration being provided for the entire train only by the brake system of the truck-tractor. The equation of motion for the trailer train in braking by the brake system of the truck-tractor can be obtained by projecting the for-

ces acting on the trailer train (Fig. 12.2) onto the surface of the road:

$$P_{br1} + P_{br2} + P_{rdt1} + P_{rdt2} + P_{rd.\,tra} - P_{in.\,t} - P_{in.\,tra} = 0 \quad (12.4)$$

where the subscripts "t" refer to the truck-tractor and "tra", to the trailer.

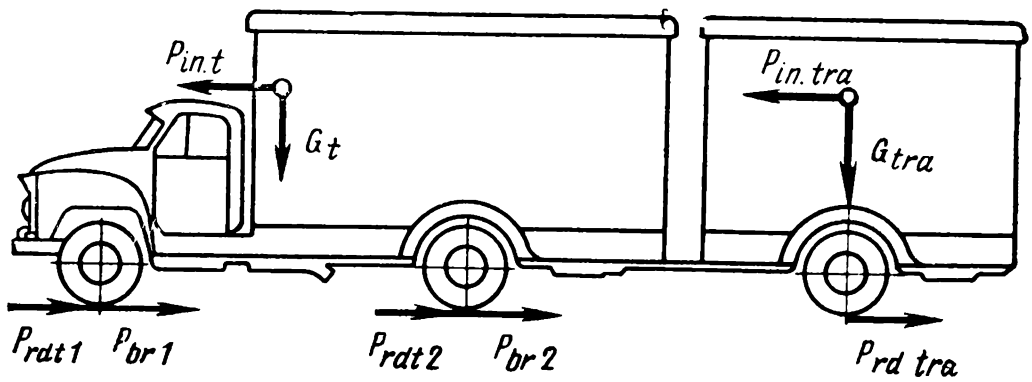


Fig. 12.2. Forces acting on a trailer train in braking

Let us use equality (12.4) to find the total braking force

$$P_{br} = \delta_{t\text{-}t}(M_t + M_{tra})\, j_{t\text{-}t} - \psi g (M_t + M_{tra}) \quad (12.5)$$

where $\delta_{t\text{-}t}$ = coefficient accounting for the rotating masses of the trailer train

$j_{t\text{-}t}$ = deceleration of the trailer train, m/s^2

M_t and M_{tra} = masses of the truck-tractor and trailer

The maximum braking force ensured by the truck-tractor

$$P_{br} = \varphi_x G_t$$

It follows from the two latter equations that

$$\varphi_x G_t = j_{t\text{-}t}\delta_{t\text{-}t} M_{t\text{-}t} - \psi G_{t\text{-}t}$$

Solving the found equation with respect to deceleration $j_{t\text{-}t}$ we get (in m/s^2)

$$j_{t\text{-}t} = \frac{\varphi_x G_t + \psi G_{t\text{-}t}}{\delta_{t\text{-}t} M_{t\text{-}t}} \quad (12.6)$$

If the deceleration $j_{t\text{-}t}$ is known it is possible to find the braking distance and time of the trailer train.

12.2. Fuel Economy of a Trailer Train

The motor vehicle incorporated in a trailer train requires more fuel per unit run but the increase in fuel consumption is not proportional to the growth in the forces of resistance to motion, since when towing trailers or semitrailers the degree of engine power

utilization is higher than in the case of a single motor vehicle, and this reduces the specific effective fuel consumption g_e.

Besides, fuel consumption per unit of transportation work (or per unit of the mass of the shipped load) noticeably drops, which diminishes the transportation cost. The fuel saving may amount to 15-20 per cent per ton-km with trailers on hard roads without steep and long ascents.

When rating the fuel consumption for a trailer train, use is made of the same equation as for a single vehicle [see equation (10.14)]. Since trailer trains usually carry loads over large distances, only the first two terms of this equation are utilized. For this reason the fuel consumption norm k_{f1} for 100 km for a trailer train is increased by 2.5 litres per ton of the trailer mass as compared with the single vehicle norm in the case of truck-tractors having carburettor engines, and by 1.5 litres in the case of diesel-powered truck-tractors. The norm k_{f2} of consumption per unit of transportation work remains the same as for a single vehicle.

Chapter XIII STABILITY OF A MOTOR VEHICLE

13.1. Lateral Stability Characteristics

The loss of stability causes a motor vehicle to topple or slip. Distinction is made between *longitudinal* and *lateral stability* depending on the toppling and slipping direction. The disturbance of lateral stability is more likely to occur and therefore especially dangerous. It takes place under the action of the centrifugal force, the transverse component of the vehicle force of gravity, the force of a cross wind and also due to wheel shocks from a rough road.

The lateral stability characteristics of a motor vehicle are assessed from *maximum possible speeds of motion over a circumference and the angle of transverse gradient of a road* (hill-side). This can also be determined from the transverse slipping conditions of the wheels (skidding) and toppling. Four features are considered in the assessment of lateral stability characteristics:

v_s — maximum (critical) speed of motion round a curve corresponding to the moment the motor vehicle begins to skid, m/s;

v_0 — maximum (critical) speed of motion round a curve corresponding to the moment the motor vehicle begins to topple, m/s;

β_s — maximum (critical) angle of hill-side which corresponds to the moment the wheels begin to slip transversely, degrees;

β_0 — maximum (critical) angle of hill-side which corresponds to the moment the motor vehicle begins to topple, degrees.

When the critical speed is experimentally determined the driver abruptly turns the steering wheel of the motor vehicle running straight on a horizontal area so that the vehicle is made to travel as fast as possible round a curve of a definite radius.

In curvilinear motion, the inner wheels with respect to the centre of turn are unloaded under the action of the centrifugal force, and the outer ones loaded. To stop the vehicle being tested from toppling, an additional outer wheel *1* is attached to it on one side (Fig. 13.1*a*). If during the test the motor vehicle loses its stability and tends to topple this wheel will contact the road and prevent the toppling. At the moment the wheels begin to slip or leave the road, the observers determine the speed v_s or v_0 with the aid of instruments.

To find the hill-side critical angle, the motor vehicle is mounted on platform *2* (Fig. 13.1) one side of which is lifted by means of pulley block *3* or jacks. As the vehicle begins to topple (or slip),

the platform inclination angle is measured with a bevel protractor.

During the theoretical analysis of factors affecting lateral stability of a motor vehicle, the magnitude of the transverse force which causes the vehicle to skid or topple should be known. When a motor vehicle moves on corner such a force is the centrifugal force. Let us consult the diagram in Fig. 13.2 to estimate this force. For simplicity we accept that the motor vehicle is a plane

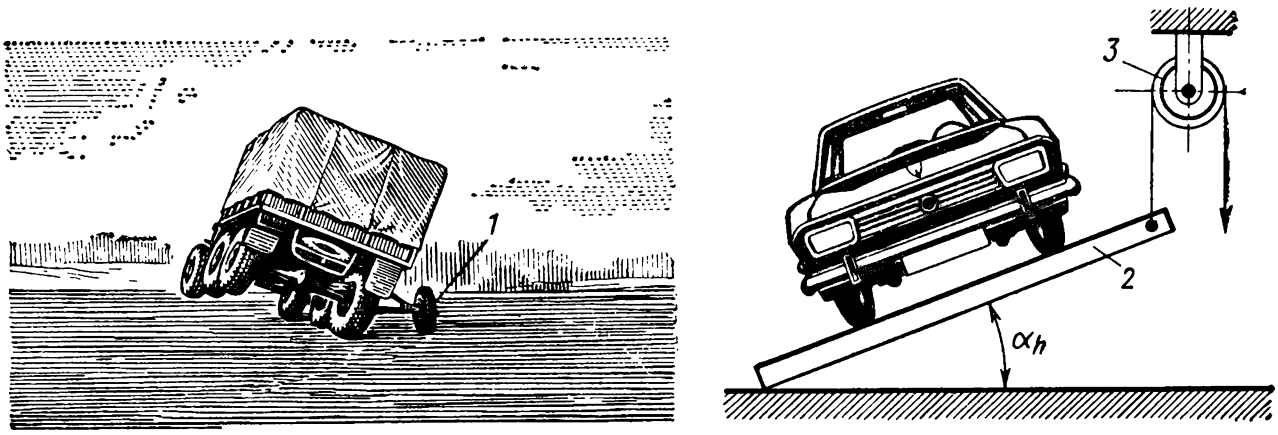


Fig. 13.1. Testing a motor vehicle for lateral stability (*a*) on corner; (*b*) on a hill-side

figure moving on a level road, and the tyres are not deformed in the transverse direction.

In the section *1-2*, the motor vehicle is in straight motion and its steerable wheels are in a neutral position. In the section *2-3*, the driver turns the wheels and the vehicle moves over a curve with a variable radius, i.e., over the first transition curve. In the section *3-4*, the position of the steerable wheels, turned through a certain angle θ, remains constant, and the radius R of the path of motion of the middle point on the rear axle is constant. In the section *4-5*, i.e., over the second transition curve, the driver turns the wheels in the opposite direction and the radius R gradually increases. In the section *5-6*, the motor vehicle regains its straight motion.

During uniform motion over an arc of constant radius the centrifugal force (in N) is

$$P_c = M_a \omega^2 \rho \qquad (13.1)$$

where M_a = mass of the automobile, kg
ω = angular velocity on corner, rad/s
ρ = radius of curvature of the gravity centre of the automobile (see Fig. 13.2), m

At the same time

$$\omega = \frac{v}{R}; \quad \rho = \frac{R}{\cos\gamma}; \quad R = \frac{L}{\tan\theta} \approx \frac{L}{\theta} \tag{13.2}$$

where γ = angle between the radius ρ and the extension of the axis of the rear wheels

θ = angle between the longitudinal axis of the motor vehicle and the speed vector of the middle point on the front axle. This angle is approximately equal to half the sum of the angles of turn of the steerable wheels (see Sec. 14.4)

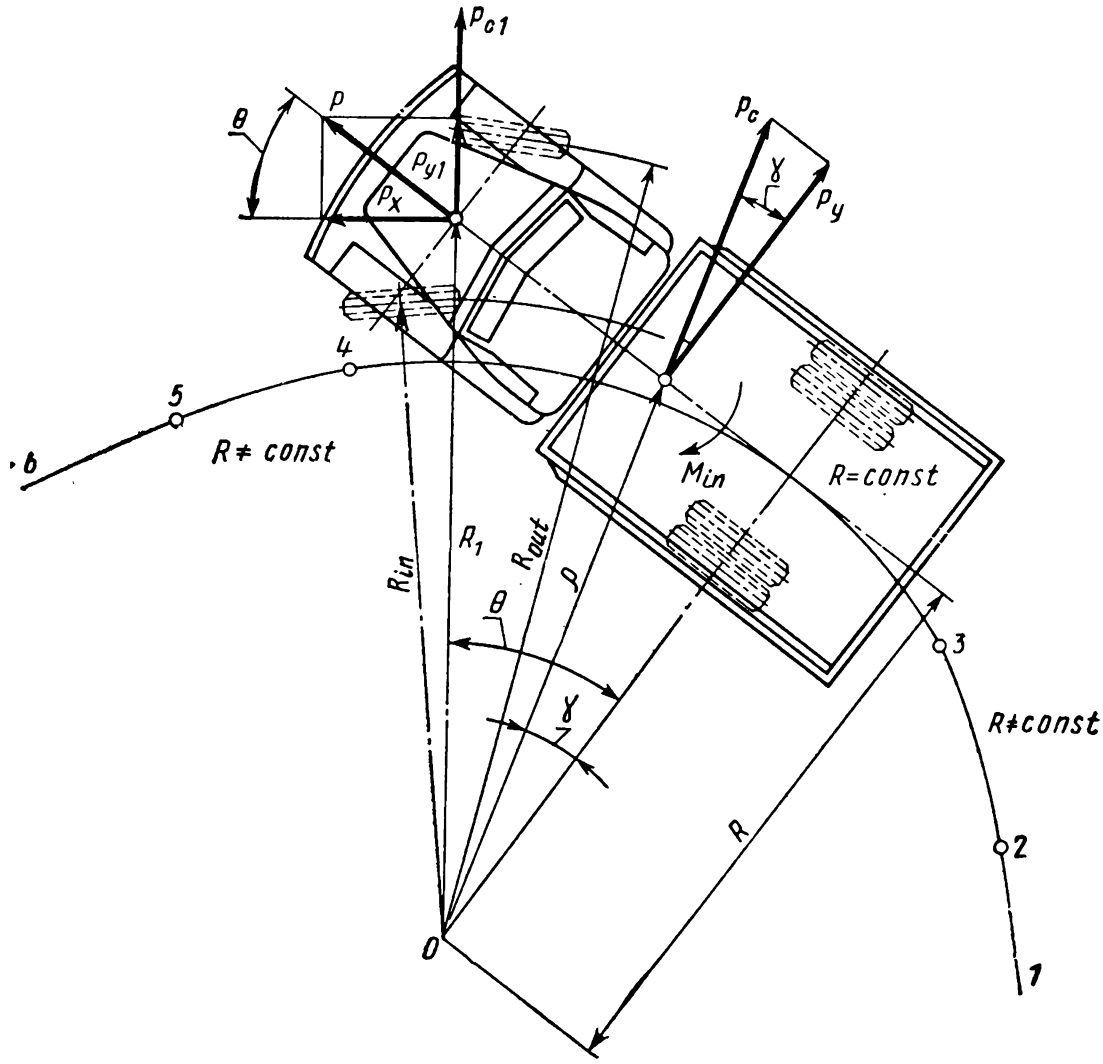


Fig. 13.2. Cornering

The loss of stability is especially dangerous when a motor vehicle runs at a high speed and its motion is almost linear. In this case the angle θ is comparatively small and it may be assumed that $\tan\theta \approx \theta$ rad.

Thus, the centrifugal force acting on the motor vehicle in uniform motion over a circumferential arc will be

$$P_c = \frac{M_a v^2}{R\cos\gamma} \approx \frac{M_a v^2 \theta}{L\cos\gamma} \tag{13.3}$$

The transverse component of the centrifugal force (in N)

$$P_y = P_c \cos \gamma \approx \frac{M_a v^2 \theta}{L} \tag{13.4}$$

In motion over the transition curves the motor vehicle is also acted upon by the force caused by the change in the path curvature. The transverse component of this force (in N)

$$P'_y = \frac{M_a v b}{L} \omega_{s.w} \tag{13.5}$$

where b = distance from the automobile centre of gravity to the rear axle, m

$\omega_{s.w}$ = angular velocity of turning the steerable wheels, rad/s

The total centrifugal force acting on the motor vehicle as the steerable wheels are turned is

$$P_{tot} = P_y + P'_y = \frac{M_a}{L} (v^2 \theta + v b \omega_{s.w}) \tag{13.6}$$

The force P_y acting on the motor vehicle during curvilinear motion is proportional to the square of the vehicle speed and the angle θ. The force P'_y acts only when the front wheels are steered and grows with an increase of their angular velocity and the speed of the motor vehicle. As the vehicle turns, the speed $\omega_{s.w}$ is positive, and the force P'_y being added to the force P_y augments the toppling or skidding hazard. When the turn is completed and the motor vehicle again resumes a straight run the velocity $\omega_{s.w}$ is negative and it may continue to move at a higher speed without losing its stability. Practically speaking, the force P'_y affects the stability only at the initial moment of time (when a turn commences) and at the final moment (when straight run is resumed) when it is commensurable with the force P_y. The effect of the force P'_y is negligible in most cases on other sections of the path.

Curvilinear motion also generates an inertia moment acting in a horizontal plane, but usually the effect of this moment on stability is small and it may be disregarded (see moment M_{in} in Fig. 13.2).

Example. Find the transverse component of the centrifugal force acting on a car when it runs over a transition curve at a speed of 10 m/s at the moment of time $t = 0$ and $t = 2$ s if $M_a = 1790$ kg, angular velocity $\omega_{s.w} = 0.05$ rad/s and $b = 1.3$ m.

At the initial moment of motion over the transition curve, i.e., at $t = 0$, the angle $\theta = 0$ and therefore the first addend in the right side of formula (13.6) is also equal to zero.

Hence,

$$P_{tot} = P'_y = \frac{M_a v b}{L} \omega_{s.w} = \frac{1790 \times 10 \times 1.3}{2.7} 0.05 = 430 \text{ N}$$

Two seconds after the turn begins, the steerable wheels will be turned by an angle

$$\theta = \omega_{s.w} t = 0.05 \times 2 = 0.1 \text{ rad}$$

$$P_{tot} = P_y + P'_y = \frac{1790}{2.7}(100 \times 0.1 + 10 \times 1.3 \times 0.05) = 6640 + 430 = 7070 \text{ N}$$

Therefore, two seconds after the turn begins, the force P_y, absent at the initial moment, becomes equal to 6640 N, i.e., it exceeds the force P'_y (430 N) by fifteen times. Thus it is possible to discard the force P'_y in the calculations and take into account only the force P_y.

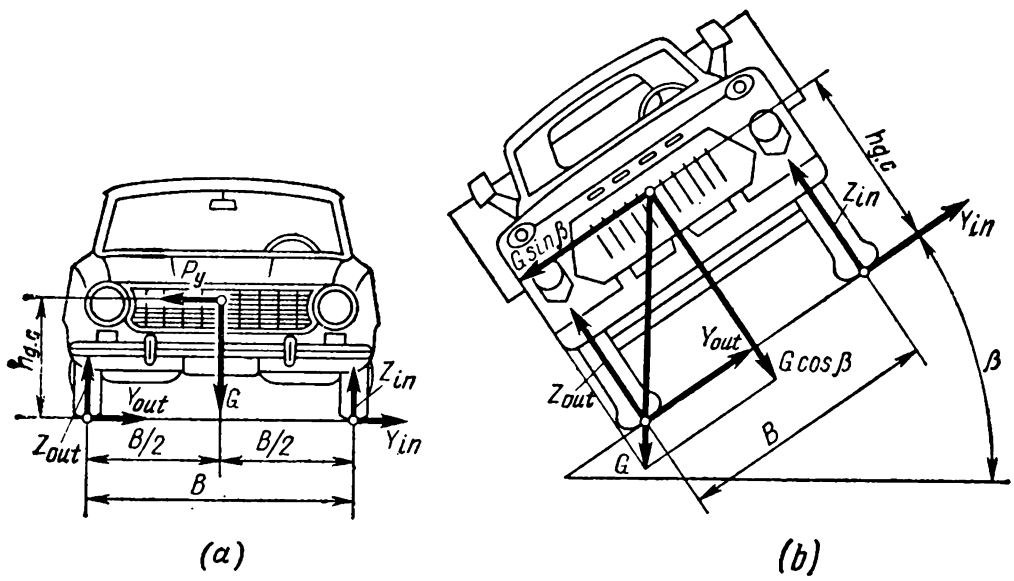


Fig. 13.3. Schemes for calculating
(*a*) critical speed; (*b*) critical angle of hill-side

Let us estimate the critical speeds of the motor vehicle as to the conditions of toppling and skidding (Fig. 13.3*a*). When it turns under the action of the centrifugal force P_y, the vehicle may topple with respect to the axis passing through the contact centres of the tyres of the outer wheels with the road. Let us write the equation of force moments relative to this axis:

$$G0.5B - P_y h_{g.c} = Z_{in} B \qquad (13.7)$$

where Z_{in} is the sum of normal reactions of the road acting on the inner wheels of the vehicle in N.

At the moment toppling begins, the inner wheels will be detached from the road, and the reactions Z_{in} will become equal to zero. Then

$$0.5GB = P_y h_{g.c} \qquad (13.8)$$

Substituting the force P_y for its value from formula (13.4) an expression can be obtained for the *critical speed for the toppling conditions* (in m/s):

$$v_0 = \sqrt{\frac{gGL}{2h_{g.c}\theta}} \approx \sqrt{\frac{gBR}{2h_{g.c}}} \qquad (13.9)$$

The action of the force P_y may also cause the tyres to slip on the road in a transverse direction (skidding). The sum of the transverse reactions Y_{in} and Y_{out} of the road is equal, in this case, to the sum of the gripping forces of all the tyres with the road:

$$Y_{in} + Y_{out} = \varphi_y G;\ Y_{in} + Y_{out} = P_y = \frac{M_a v^2 \theta}{L}$$

Hence, the *critical speed for the skidding conditions* (in m/s)

$$v_s = \sqrt{\frac{\varphi_y g L}{\theta}} \approx \sqrt{\varphi_y g R} \tag{13.10}$$

Example. Find the critical speeds v_0 and v_s of a motor vehicle moving over an arc of radius 30 m; $B = 1.47$ m; $h_{g.c} = 0.6$ m; $\varphi_y = 0.8$.

The angle of turn of the steerable wheels

$$\theta \approx \frac{L}{R} = \frac{2.7}{30} = 0.09 \text{ rad}$$

The critical speed for the toppling conditions

$$v_0 = \sqrt{\frac{gBL}{2h_{g.c}\theta}} = \sqrt{\frac{9.81 \times 1.47 \times 2.7}{2 \times 0.6 \times 0.09}} = 19 \text{ m/s}$$

The critical speed for the skidding conditions

$$v_s = \sqrt{\frac{\varphi_y g L}{\theta}} = \sqrt{\frac{0.8 \times 9.81 \times 2.7}{0.09}} \approx 13.4 \text{ m/s}$$

The motor vehicle may also lose its lateral stability on a straight run if the driver abruptly turns the wheels even through a small angle. The centrifugal force generated in this case may rapidly reach the value of the grip of the tyres with the road and cause skidding.

Let us determine the space of time during which the centrifugal force increases to the dangerous limit, assuming that the driver turns the wheels at a constant rate. At the moment skidding arises, the force $P_{tot} = P_{gr}$ or

$$\frac{M_a \omega_{s.w}}{L}(v^2 t + vb) = \varphi_y G_a \tag{13.11}$$

Hence, the time (in s)

$$t = \frac{1}{v}\left(\frac{\varphi_y g L}{v \omega_{s.w}} - b\right) \tag{13.12}$$

If the speed of the motor vehicle is high and the grip coefficient small a sharp turn of the steerable wheels will cause skidding within a very brief interval of time. In specially unfavourable conditions, this time may be less than the driver's reaction time and he may fail to prevent skidding.

Example. Determine the time during which skidding is likely to appear if a motor vehicle moves on a dry asphalt-concrete surface ($\varphi_y = 0.8$) and on an icy road ($\varphi_y = 0.2$) at a speed of 10 m/s and the driver sharply turns the steerable wheels (angular velocity $\omega_{s.w} = 0.2$ rad/s).

During motion on the dry surface

$$t = \frac{1}{v}\left(\frac{\varphi_y g L}{v\omega_{s.w}} - b\right) = \frac{1}{10}\left(\frac{0.8 \times 9.81 \times 2.7}{10 \times 0.2} - 1.3\right) = 0.93 \text{ s}$$

and on the icy road

$$t = \frac{1}{10}\left(\frac{9.81 \times 2.7 \times 0.2}{10 \times 0.2} - 1.3\right) = 0.13 \text{ s}$$

When a motor vehicle runs on a transverse hill-side stability may be lost due to the action of the transverse component of the force of gravity of the motor vehicle equal to $G \sin \beta$ (Fig. 13.3*b*). Let us write the moment equation for all forces with respect to the axis passing through the tyre contact centres of the outer wheels with the road:

$$z_{in}B + G \sin \beta h_{g.c} = G \cos \beta 0.5\, B \qquad (13.13)$$

At the moment toppling begins, the reaction $Z_{in} = 0$. Thus, *the hill-side critical angle for the toppling conditions*

$$\beta_0 = \arctan\left(\frac{B}{2h_{g.c}}\right) \qquad (13.14)$$

The hill-side critical angle as to skidding can be determined by projecting all forces onto the plane of the road

$$G \sin \beta = Y_{in} + Y_{out} \qquad (13.15)$$

At the same time, by the conditions of tyre grip with the road the sum of the transverse reactions at the moment skidding begins

$$Y_{in} + Y_{out} = \varphi_y G \cos \beta \qquad (13.16)$$

Hence, *the hill-side critical angle for the skidding conditions*

$$\beta_s = \arctan \varphi_y \qquad (13.17)$$

It has been mentioned above that a motor vehicle may lose its stability due to a curvature of the road or its transverse inclination. Both factors may also act simultaneously. Figure 13.4 shows a vehicle B which runs on the inner edge of the road, and a vehicle A moving on its outer edge. Let us resolve the vehicle force of gravity G into the force G_z perpendicular to the road and the force G_y parallel with it. The transverse component P_y of the centrifugal force is to be resolved along the same directions into the forces P_{yz} and P_{yy}. The forces G_y and P_{yy} of vehicle A are added because they are directed similarly. The force P_{yz} is directed oppositely to the force G_z and this diminishes the tyre road grip

force. The probability of losing stability is greater for vehicle *A* than for vehicle *B* as the forces G_y and P_{yy} are subtracted, since they are directed oppositely and, conversely, the forces G_z and P_{yz} are summed because they are in the same direction. Therefore, a left-hand turn is especially dangerous on a crowned road (right-hand traffic).

To ensure safe traffic over curves of small radii, the road and its sides are sloped transversely towards the centre of the curve. If the road is sloped in this manner the components G_y and P_{yy}

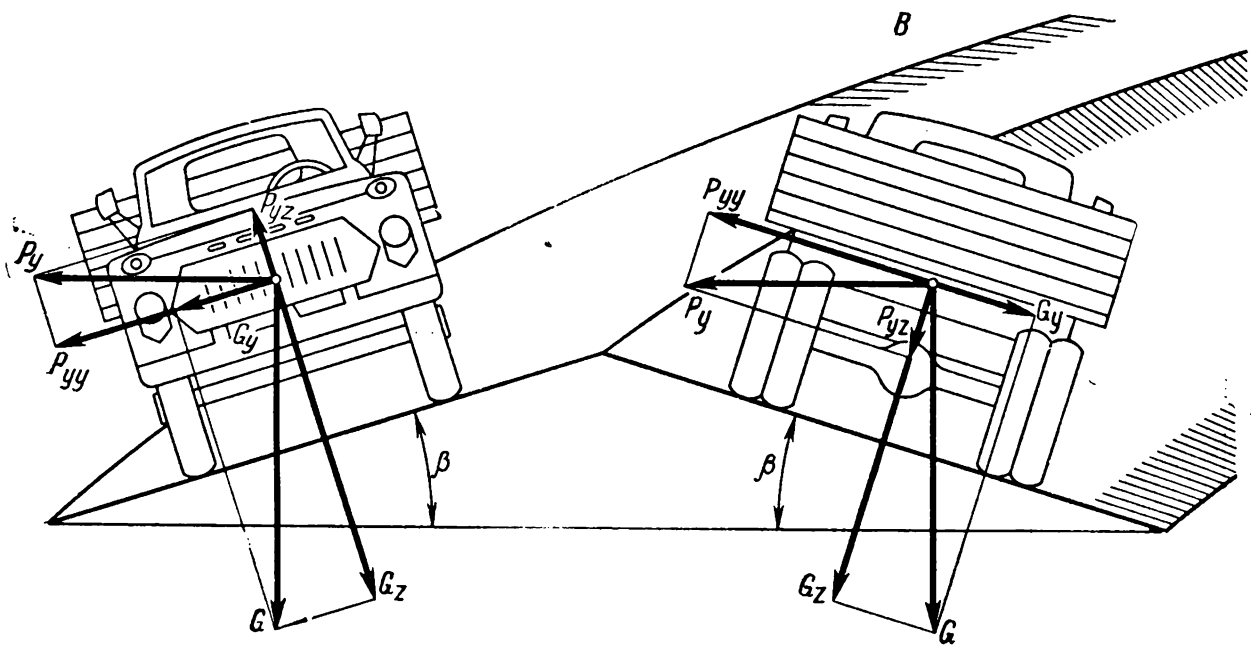


Fig. 13.4. Right-hand and left-hand cornering

are subtracted and the components G_z and P_{yz} are added irrespective of the direction of motion; this improves the lateral stability of motor vehicles. With a smaller curve radius the slope gradient is increased.

The lateral stability considered above assumed that in the case of skidding both axles begin to slip simultaneously in the transverse direction. In actual fact, this seldom occurs. Usually, the wheels of one axle begin to slip, in view of which the stability of one of the axles and not of the entire vehicle has to be considered.

In conformity with formula (7.23) the following condition is to be observed for a wheel to roll without slipping:

$$Y \leqslant \sqrt{\varphi^2 Z^2 - X^2} \qquad (13.18)$$

Thus, the greater the gripping force and the smaller the tangential road reaction, the higher the transverse force that can be applied to the wheel without causing it to slip. The driven wheel is most stable in the transverse direction. In this wheel the tan-

gential reaction which represents the force of resistance to rolling is small as compared with the force φZ. A wheel loaded with a tractive or braking force resists skidding worse than the driven wheel. If the tangential reaction reaches the magnitude of the gripping force it is enough to apply a small transverse force to the wheel for lateral slipping to occur.

Figure 13.5*a* shows a motor vehicle in which the front wheels move translationally with a speed of v_1, and the rear axle moving at a speed of v_1 slips at v_2 due to transverse skidding. As a result, the rear axle moves at a speed of v_3, which causes the motor vehicle to turn around the centre O. The transverse component of the generated centrifugal force P_c acts in the direction of the rear axle slipping and tends, therefore, to intensify the skidding. Increased skidding causes a further growth in the centrifugal force, and the skidding progresses. For this reason, the rear axle skidding is more dangerous than that of the front axle (Fig. 13.5*b*) when the transverse component of the force P_c is applied in the direction opposite to the lateral slipping speed v_2. This automatically stops the slipping of the front wheels, and the motor vehicle does not lose its stability. For all that, even a short-time lateral slip of the front wheels may cause loss of control (see Chapter XIV).

To eliminate the rear axle skidding, it is necessary to reduce the tangential reaction on the driving wheels by releasing the brakes or slightly closing the throttle, and turning the front wheels in the direction of skid. If the front wheels occupied a neutral position during skidding and the centre of turn was at the point O (Fig. 13.5*c*), it will be displaced to the point O_1 after the front wheels are turned. In this case the radius of turn will increase and reduce the magnitude of the centrifugal force.

If the front wheels are turned towards the side of skidding through an angle at which the speed vectors of the rear and front axles are parallel, the motor vehicle will no longer turn and will begin to move translationally in the direction of these vectors. Finally, when the front wheels are turned through a greater angle the centre of turn will be located on the opposite side of the motor vehicle. In this case the transverse component of the centrifugal force will be directed to the side opposite to skidding which will stop.

When a motor vehicle moves on a slippery road, skidding may occur instantaneously and the driver must react forthwith. The steering wheel must be sharply turned to eliminate the skidding. The angular velocity of turning the front wheels may be several times larger than during ordinary driving. This produces a sharp increase of the second component of the centrifugal force (force P'_y) caused by the change in the path curvature. Turning the front

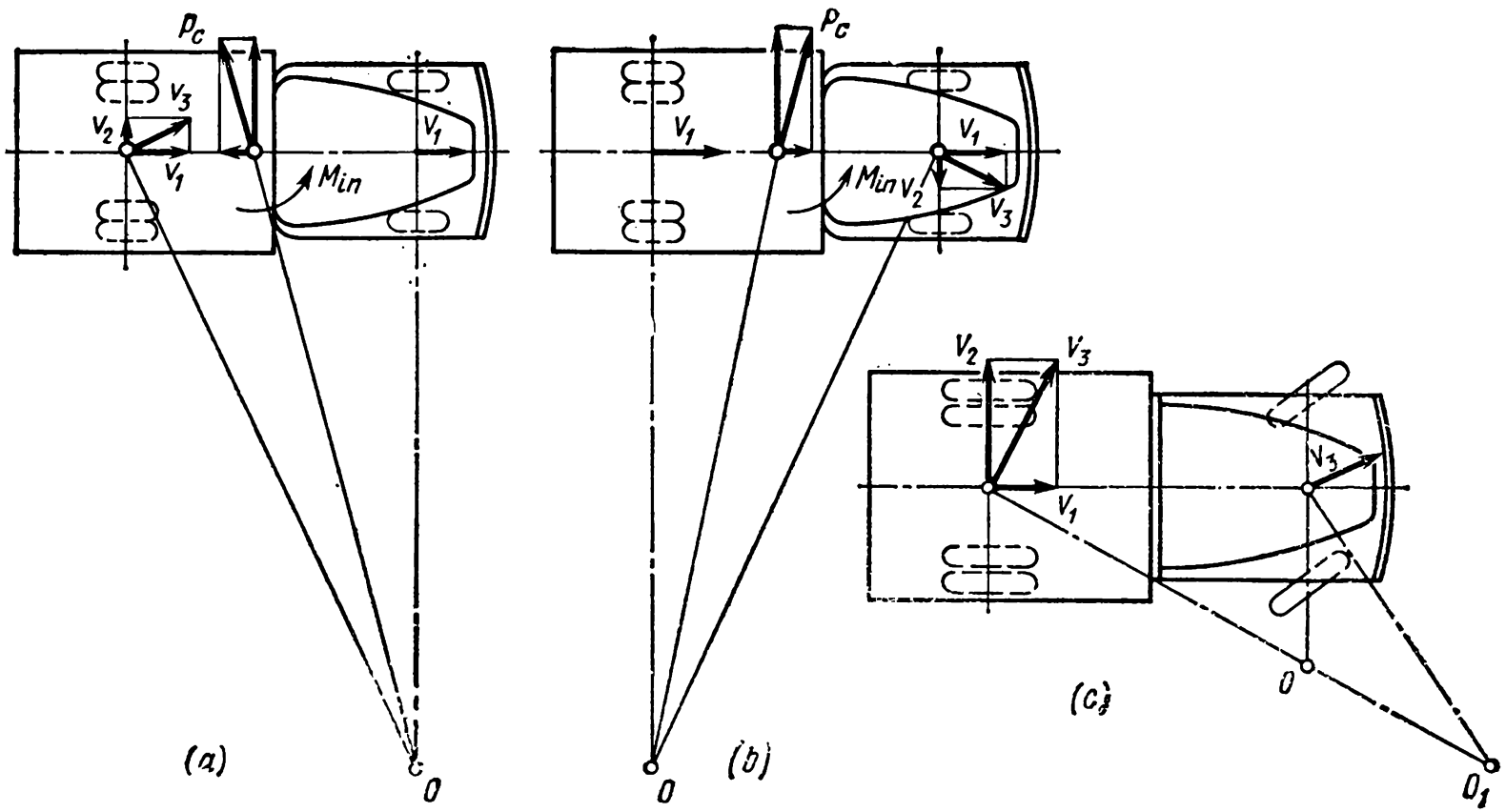


Fig. 13.5. Skidding of motor vehicle axles
(*a*) rear; (*b*) front; (*c*) elimination of skidding

wheels through an excessively large angle may cause the rear wheels to slip in the other direction and the vehicle to move in the direction which corresponds to the new position of the steerable wheels. Therefore they should be returned to the neutral position as soon as skidding is over.

The loss of stability can be prevented and safe motion ensured if the speed of the motor vehicle is reduced before a turn especially on a moist and slippery road.

13.2. Body Lateral Rolling

It has been accepted when determining the stability characteristics that a motor vehicle is a solid all points of which move at the same speed. In actual fact, a motor vehicle is a complex

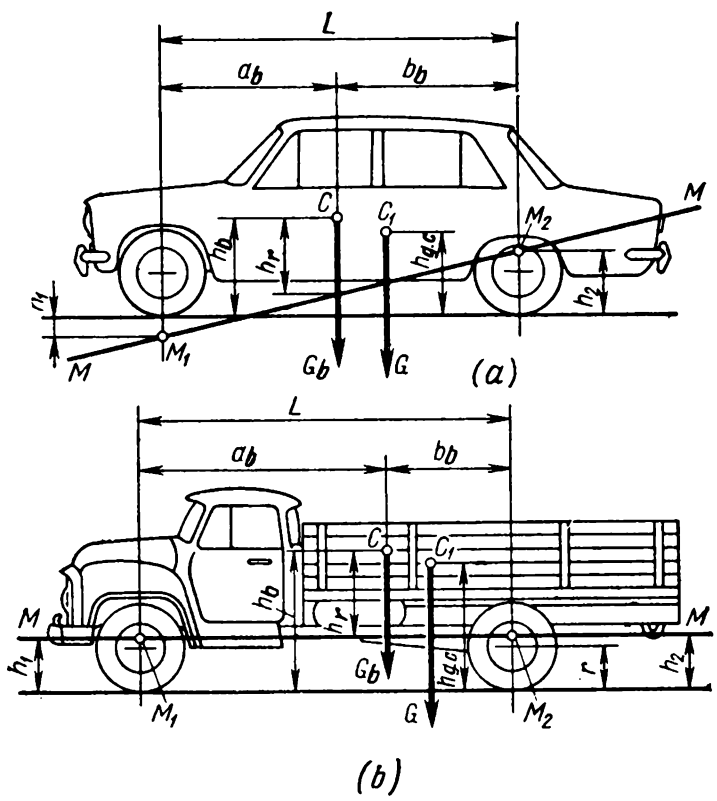


Fig. 13.6. Arrangement of the body roll centres and axis
(*a*) car; (*b*) truck

system of masses jointed by hinges or elastic elements. Distinction may be made between two basic groups of masses: the spring-mounted parts (body) whose weight is taken by the suspension, and the parts without springs (wheels, axles) whose weight is carried by the tyres. The centre of gravity C of the spring-mounted masses is arranged at a distance of a_b from the front axle (Fig. 13.6), at a distance of b_b from the rear axle and at a height h_b from the road surface. It does not coincide with the centre of gravity C_1 of the motor vehicle. The point C is located

above the point C_1 and therefore $h_b > h_{g.c.}$. The mass of non-springy parts of the front axle is usually slightly less than the mass of those of the rear axle, in view of which the point C_1 is displaced backwards with respect to the point C ($b < b_b$).

Under the action of the transverse force, the tyres and the elastic elements of the suspension are relieved on one side and

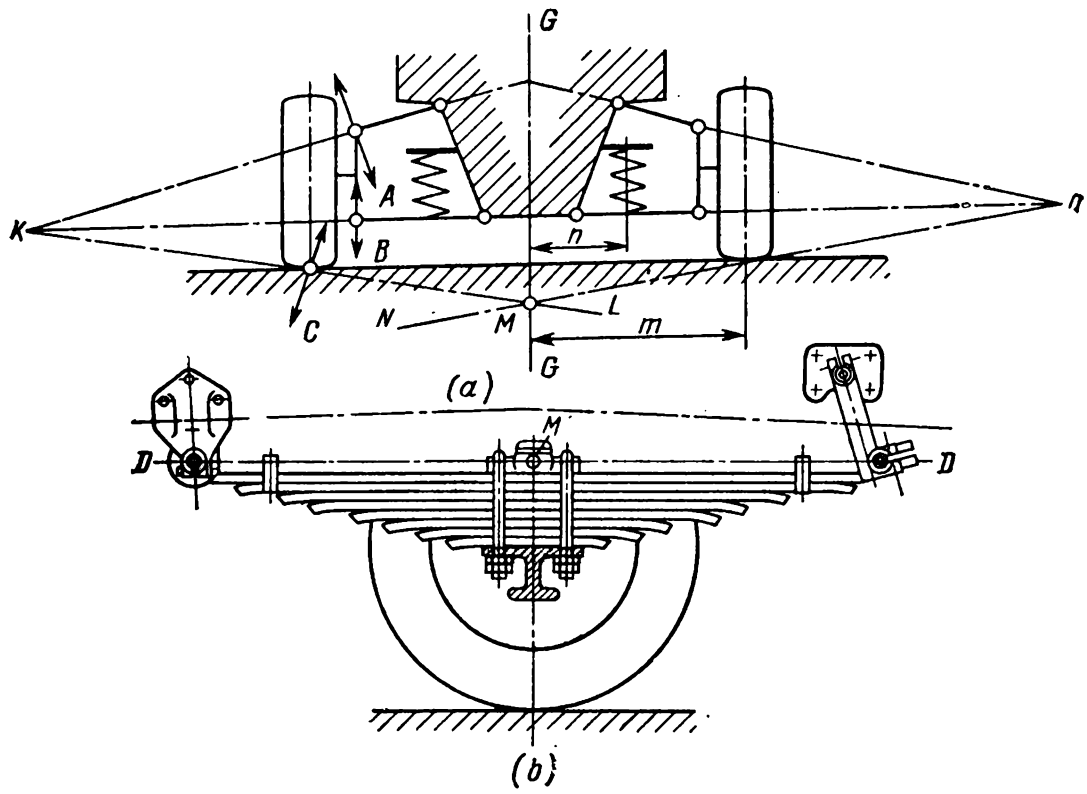


Fig. 13.7. Determining the roll centre with various types of suspension
(*a*) parallel-arm; (*b*) leaf-spring

loaded on the other. As a result, the body of the motor vehicle is inclined and turns in the transverse direction.

The roll centre (front or rear) is the point with respect to which the transverse section of the vehicle passing through the front or rear axle moves.

The roll axis is the straight line *MM* with respect to which the body being inclined turns transversely. The roll axis passes through the roll centres of the front and rear portions of the body.

Let us find the position of the roll centre in the case of a parallel-arm suspension. For this purpose, we consider the diagram in Fig. 13.7*a*. With a stationary body, vertical oscillations of the right-hand wheel cause the points *A* and *B* (ends of the upper and lower arms) to move in the directions perpendicular to the arms. The instantaneous centre of speeds of the wheel is located at the point *K* of intersection of the lines which are the extensions

of the arms. The point C of the wheel, where it contacts the road, moves in this case square with the line KL.

If the wheel does not move and the body is inclined the roll centre should also lie on the line KL. The right- and left-hand halves of the suspension are symmetric with respect to the vertical plane GG and, therefore, the roll centre M lies at the point of intersection of the lines KL and ON, i.e., in the plane GG. In our case it is slightly below the road surface.

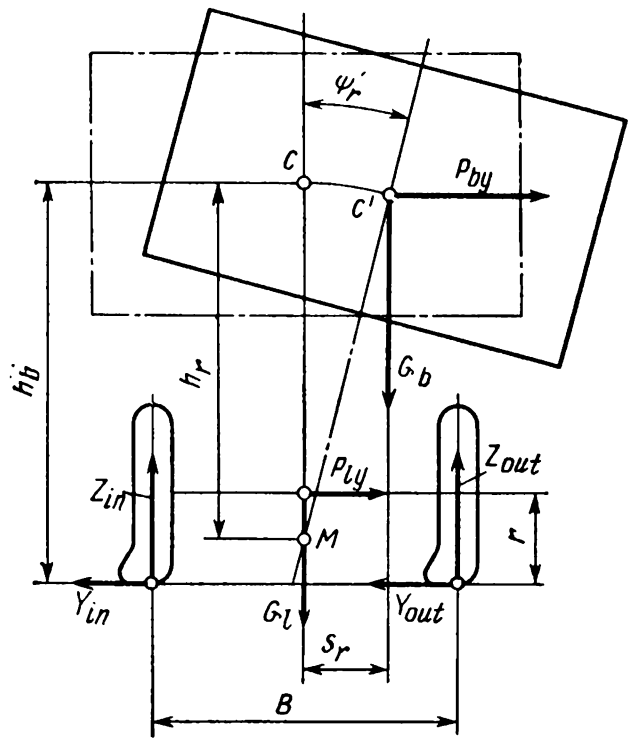


Fig. 13.8. Determining the body roll angle

A suspension which allows the wheel to run strictly parallel may be regarded as an arm suspension with infinitely long arms. With such suspension, the roll centre lies on the surface of the road. If a motor vehicle is provided with an unsplit axle and a suspension with longitudinal leaf springs the roll centre will lie in the plane DD passing through the eyes of the springs (Fig. 13.7*b*).

In cars supported by a front independent and a rear spring suspension, the roll axis is inclined towards the road (see Fig. 13.6*a*). In trucks and buses with spring suspensions, the roll axis lies parallel to the road approximately at a height equal to the radius of the wheel (see Fig. 13.6*b*) and $h_1 \approx h_2$.

The transverse force P_{by} applied to the centre of gravity of the body acts relative to the roll axis on the roll arm h_r and builds up a moment which causes the body to incline transversely.

The roll arm (see Fig. 13.6) is

$$h_r = h_b - \frac{a_b h_2 - b_b h_1}{L} \qquad (13.19)$$

where h_1 and h_2 = distances from the road surface to the front and rear roll centres, respectively, m

Let us determine the roll angle ψ_r when a motor vehicle makes a turn on a level road (Fig. 13.8). The force of gravity G_b and the centrifugal force P_{by} are applied to the centre of gravity of the spring-mounted masses. The point of application of the force of gravity G_l and the centrifugal force P_{ly} of the springless masses is arranged at a height approximately equal to the radius of the

wheel. The force P_{by} turns the body with respect to the roll axis through an angle of ψ_r and point C is displaced to point C'. The wheels on the inner side are acted upon by the reactions of the road Z_{in} and Y_{in}, and the wheels on the outer side, by the reactions Z_{out} and Y_{out}. The angle ψ_r does not usually exceed 10° and for this reason it may be assumed that $h_b \approx$ const.

Now we consider the equilibrium of a body acted upon by the forces G_b, P_{by} and the reactions of the suspension during a transverse roll. Taking with small values of the angle ψ_r the transverse displacement of the centre of gravity (in m)

$$s_r = h_r \tan \psi_r \approx h_r \psi_r \tag{13.20}$$

the roll angle (in rad) may be determined as follows

$$\psi_r = \frac{P_{by} h_r}{c_{ang.a} - G_b h_r} \tag{13.21}$$

where $c_{ang.\,a}$ is the angular rigidity of suspension equal to the ratio of the moment which causes the body to roll to the roll angle, N·m/rad.

The roll angle increases with a higher transverse force and longer roll arm and diminishes when the angular rigidity of the suspension increases. If the front spring suspension is replaced by an independent suspension the roll arm increases (see Fig. 13.6*a*) which appreciably intensifies the transverse roll of the body. A stabilizer of lateral stability is utilized to diminish the roll and increase thereby the angular rigidity of the suspension.

Example. Determine the roll angle of a vehicle body in uniform motion over a curvature with a radius of 40 m at a speed of 10 m/s; the weight of the body $G_b = 15{,}000$ N; mass of the body $M_b = 1500$ kg; $h_r = 0.5$ m; $c_{ang.\,a} = 30{,}000$ N·m/rad.

The transverse force of the spring-mounted masses

$$P_{by} = \frac{M_b v^2}{R} = \frac{1500 \times 10^2}{40} = 3750 \text{ N}$$

The body roll angle

$$\psi_r = \frac{P_{by} h_r}{c_{ang.\,a} - G_b h_r} = \frac{3750 \times 0.5}{30{,}000 - 15{,}000 \times 0.5} \approx 0.085 \text{ rad} \approx 5°$$

Due to rolling, the centre of gravity of the body is displaced in the direction of action of the transverse force and the outer wheels are loaded additionally and the inner wheels are relieved of load still more. For this reason, a body roll increases the probability of the motor vehicle toppling. If, when determining the speed v_0 and the angle β_0, account is also taken of the body roll, their values will prove to be 10-15 per cent less than those given by formulas (13.9) and (13.14).

13.3. Effect of Operating Factors on Lateral Stability

The loss of lateral stability is usually observed during braking, since high braking forces come into play where the tyres contact the road. Therefore, the wheels cannot take even slight transverse forces without slipping. When the wheels are locked their motion is unstable.

The stability of a motor vehicle during braking can be impaired by a nonuniform braking force distribution between the wheels

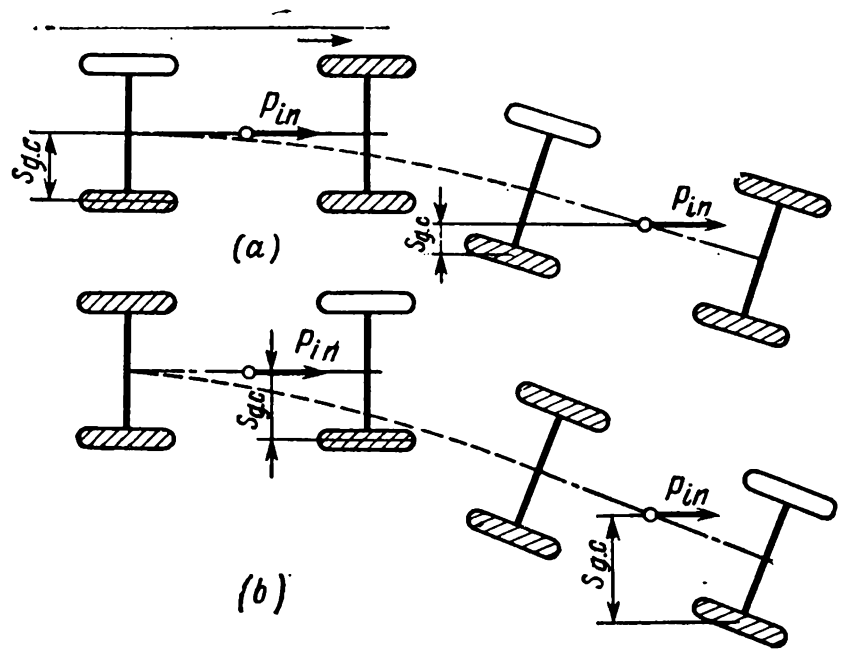


Fig. 13.9. Stability of a motor vehicle when braked:
(a) right-hand rear wheel braked; (*b*) one of the front brakes faulty

due to oiling of the friction linings or wrong adjustment of the brakes. Nonuniform distribution of the braking forces in the front wheels is, as a rule, more dangerous than in the rear wheels. If only one rear wheel, for example, the right-hand one (shown shaded in Fig. 13.9*a*), is braked and the other (left-hand) runs freely, the motor vehicle will deflect to the right from the straight path. The distance $s_{g.c}$ from the centre of gravity to the braked wheel diminishes in this case as does the turning moment built up by the force of inertia P_{in}.

When one of the front brakes is faulty (Fig. 13.9*b*) the arm $s_{g.c}$ increases during braking, which tends to deviate the motor vehicle still further to the side. For this reason, a defect of the front brakes is more dangerous than that of the rear brakes.

The same phenomenon occurs when the braked wheels on one side of the motor vehicle roll on a slippery portion of the road and on the other side, on a surface with a high φ coefficient.

During braking the lateral stability may also be disturbed because of improper stowage of load in the body. If the centre of gravity of the load does not coincide with the longitudinal axis of the vehicle, the force of inertia P_{in} generated during braking produces a turning moment on the arm $s_{g.c}$ (Fig. 13.10*a*). If in this case the front wheels are locked (shown shaded in Fig 13.10*a*) their force of adhesion with the road is less than that of the rear wheels, and therefore the moment $P_{in}s_{g.c}$ turns the motor vehicle being braked with respect to the middle point on the rear axle. In this case, the arm $s_{g.c}$ is rapidly reduced to zero and the vehicle will no longer be turned. If the rear wheels are locked (Fig. 13.10*b*) the vehicle turns relative to the middle point on the front axle, the arm $s_{g.c}$ continually increasing, which augments the skidding hazard.

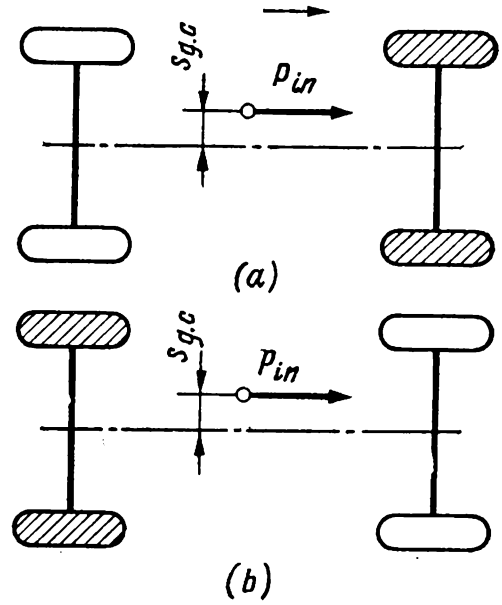


Fig. 13.10. Stability of a motor vehicle with wrong stowage
(*a*) front wheels locked; (*b*) rear wheels locked

If a motor vehicle is braked with the engine not disconnected from the transmission, the lateral stability is improved, since the differential gear divides the braking force caused by the friction in the engine and transmission equally between the rear wheels. This reduces the nonuniformity of the braking forces distribution produced by the different braking moments or grip coefficients and decreases the skidding tendency.

13.4. Longitudinal Stability

The loss of longitudinal stability may cause the motor vehicle to topple relative to the front or rear axle or slip (slide) in the longitudinal direction. Modern vehicles with their low centre of gravity are most unlikely to topple over in the longitudinal plane. Only the rear wheels may skid and cause the vehicle to slide backward down the slope, for example, when a trailer train dynamically overcomes a long steep hill. Going up such a hill, the trailer train may use the store of the kinetic energy before the ascent ends and the sum of the resistance forces proves larger than the tractive force.

In order to determine the maximum angle of gradient that can be overcome by a trailer train in uniform motion without the driving wheels of the truck-tractor skidding, we revert to the

diagram of the forces acting on the motor vehicle (see Fig. 7.11). Let us assume for simplicity that the forces P_{r1}, P_{r2}, P_{air} and also the moment M_r are equal to zero. During uniform motion the force P'_{in} is also equal to zero, and the reactions

$$X_1 = -P_{r1} \text{ and } X_2 = P_t - P_{r2} \qquad (13.22)$$

According to the accepted assumption, the reaction X_1 is equal to zero and the reaction X_2, to the tractive force P_t whose maximum value is limited by the grip of the truck-tractor driving wheels with the road and equal to the product $\varphi_x Z_2$. Then, inserting the values of the moment M_r, the forces P_{air}, P'_{in} and P_t and also the values of the forces P_g and P_{tra} equal to $G_a \sin\alpha$ and $G_{tra} \sin\alpha$ into equation (7.54) we obtain

$$Z_2 = \frac{G_a \cos\alpha a + G_a \sin\alpha (h_{g.c} - r) + G_{tra} \sin\alpha (h_{tra} - r)}{L - \varphi_x r} \qquad (13.23)$$

At the moment when slipping begins, the force of grip of the driving wheels of the truck-tractor with the road is equal to the force of resistance to ascent offered to the trailer train:

$$P_{gr} = \varphi_x Z_2 = (G_a + G_{tra}) \sin\alpha \qquad (13.24)$$

Solving equations (13.23) and (13.24) together, the expression can be obtained for the *maximum (critical) angle of gradient* at which the trailer train can move without the truck-tractor driving wheels skidding:

$$\tan\alpha_{sk} = \frac{\varphi_x G_a a}{G_a (L - \varphi_x h_{g.c}) + G_{tra} (L - \varphi_x h_{tra})} \qquad (13.25)$$

Since the weight $G_{tra} = 0$ for a single motor vehicle, then

$$\tan\alpha_{sk} = \frac{\varphi_x a}{L - \varphi_x h_{g.c}} \qquad (13.26)$$

The critical angle of gradient largely depends on the grip coefficient. At $\varphi_x = 0.3$ this angle does not exceed 4-6° for trailer trains and lies within 10-15° for single vehicles of the type 4×2. This explains why the driving wheels of the truck-tractor of a trailer train frequently skid in winter on gently sloping ascents.

For the vehicles of types 4×4 and 6×6 the force $P_{gr} = \varphi_x G_a \cos\alpha$. The critical angle of gradient which can be overcome by these vehicles without skidding and sliding backwards can be found from the expression $\tan\alpha = \varphi_x$.

A motor vehicle with all the wheels driving can climb steep ascents without losing its longitudinal stability even on wet and slippery roads (α_{sk} = 17-19°).

Chapter XIV STEERABILITY OF A MOTOR VEHICLE

14.1. Steerability Characteristics

When a motor vehicle moves, the situation in the streets and on the roads incessantly changes which calls for increased attention on the part of the driver and constant readiness to alter the nature of motion in order to prevent road accidents. For this reason, apart from stability, the design of a motor vehicle should provide for an easy and fast change in the road position of the vehicle. Vehicle control largely depends on the steering gear, suspension, and tyres.

To attain good steerability of a motor vehicles, the following requirements should be satisfied:

the steerable wheels should roll without lateral skidding as the motor vehicle is turned;

the transverse elasticity of the tyres should be such that the vehicle moves on corner over an arc of greater radius than a vehicle with tyres that are stiff in the crosswise direction;

the steering linkage should ensure correct proportion between the turn angles of the steerable wheels;

during straight run the steerable wheels should preserve their neutral position and automatically regain it when the turn is completed;

arbitrary angular oscillations of the steerable wheels should be precluded.

The degree by which any particular motor vehicle design corresponds to each of the above-listed requirements is estimated by a special index.

14.2. Rolling of the Steerable Wheels Without Slipping

Figure 13.2 shows a motor vehicle on corner. The steerable wheels are at a mean angle of θ with respect to the neutral position they take in straight motion.

The force P is applied to the front axle, its component P_x, parallel to the steerable wheels, overcomes during uniform motion the rolling resistance force $P_{r1} = fG_1$, and is equal to it in magni-

tude. Hence, the force (in N) that is the transverse component of the force P is

$$P_{y1} = P_{r1} \tan\theta = fG_1 \tan\theta \qquad (14.1)$$

The front axle is also acted upon by the centrifugal force of the vehicle front part, directed from the centre of turn along the radius R_1:

$$P_{c1} = \frac{M_1 v^2 \tan\theta}{L \cos\theta} \qquad (14.2)$$

where M_1 is the mass of the vehicle acting on the front axle.

The forces P_{y1} and P_{c1} tend to cause lateral skidding of the steerable wheels. For the wheels to roll without slipping, condition (13.18) should be satisfied:

$$Y = P_{y1} + P_{c1} \leqslant \sqrt{\varphi^2 Z^2 - X^2} \approx G_1 \sqrt{\varphi^2 - f^2} \qquad (14.3)$$

The force P_{c1} is directly proportional to the square of speed, and the wheels are therefore most probable to slip when a motor vehicle runs at a high speed.

The critical speed v_{st} as to steerability is the speed at which a vehicle can move on corner without lateral sliding of the steerable wheels. This speed can be found from expressions (14.1)-(14.3):

$$v_{st} = \sqrt{\left(\frac{\sqrt{\varphi^2 - f^2}}{\tan\theta} - f\right) gL \cos\theta} \qquad (14.4)$$

If the speed of the motor vehicle exceeds v_{st} the steerable wheels on corner will slip in the transverse direction, since the turn of the wheels does not change the direction of its motion. The critical speed v_{st} diminishes with a larger angle of θ. The smaller the radius of turn, the less should be the vehicle speed, for otherwise the steerable wheels will begin to skid laterally.

On roads with a hard surface, the coefficient f is usually much less than the coefficient φ and for this reason the vehicle can be controlled even over curves of small radii.

In motion on roads with an uneven icy covering and also on sand or snow, the values of the coefficients φ and f become closer and the difference $\varphi^2 - f^2$ diminishes, thus reducing the critical speed.

If $\varphi = \frac{f}{\cos\theta}$ the radicand is equal to zero and the motor vehicle can turn only at a very low speed.

If $\varphi < \frac{f}{\cos\theta}$ the vehicle loses its control because the speed v_{st} is an imaginary value.

The critical speed as to the conditions of steerability may be less than the skidding and toppling critical speeds. This means

that the loss of steerability is no less probable and dangerous as the loss of stability.

The transverse reaction of the road cannot occur when the front wheels fully slip, for example, when they are locked in braking. In this case, the turn of the front wheels does not change the direction of their motion and the vehicle cannot be steered.

Example. Find the critical speed for steerability of a motor vehicle if $\theta = 20°$ ($\tan 20° = 0.364$; $\cos 20° = 0.94$); $\varphi = 0.2$; $f = 0.04$.

The critical speed

$$v_{st} = \sqrt{\left(\frac{\sqrt{\varphi^2 - f^2}}{\tan\theta} - f\right) gL\cos\theta} =$$

$$= \sqrt{\left(\frac{\sqrt{0.2^2 - 0.04^2}}{0.364} - 0.04\right) 9.81 \times 2.7 \times 0.94} \approx 3.5 \text{ m/s}$$

14.3. Wheel Lateral Slip and Ability of a Motor Vehicle to Turn

Rolling of a wheel in the absence of transverse force has been discussed above. Rolling of an elastic wheel loaded with a transverse force has its own specific features, because the tyre is elastic not only in the radial but also in the crosswise direction and is deformed by the transverse force. Therefore, a wheel with a pneumatic tyre can roll on the road without sliding at a certain angle to its medium plane. This kind of rolling is known as *wheel lateral slip* and the angle formed by the speed vector of the wheel centre with its medium plane, as *slip angle.*

If the wheel is acted upon by the forces P_z and P_x only (Fig. 14.1) then, as the wheel rolls, the point B of the tyre will touch the road at the point B_1, and the point C at the point C_1, etc. The medium plane of the wheel coincides with the straight line AD_1 along which the tangential reaction X is directed.

Let a transverse force P_y be applied to the wheel. In this case, a transversely elastic tyre will bend and the medium plane of the wheel will be displaced by distance b_t with respect to the centre of contact O. As the wheel rolls, the point B of the tyre comes in contact with the road at the point B_2, the point C at C_2, etc. As a result, the wheel will run in the direction AD_2. In this case, the medium plane of the wheel will be arranged at an angle of $\delta_{l.s}$ to the direction of motion. The medium line of contact coinciding approximately with the direction of motion will also be located at an angle $\delta_{l.s}$ to the medium plane of the wheel. Thus, the wheel will roll with lateral slip whose angle is equal to $\delta_{l.s}$.

During the slip, the elements of the tyre deformed less in the transverse direction (section *kk*) than in the rear (section *nn*)

approach the front portion of the contact. Thus, the elementary transverse reactions are less in the front portion of the contact than in the rear and the diagram of the elementary transverse reactions is close to a triangle in form. The resultant Y equal in magnitude to the force P_y becomes displaced at a distance of c_t with respect to the centre of contact. The reaction X is also displaced at a distance of b_t. Due to lateral slip, the projection of the

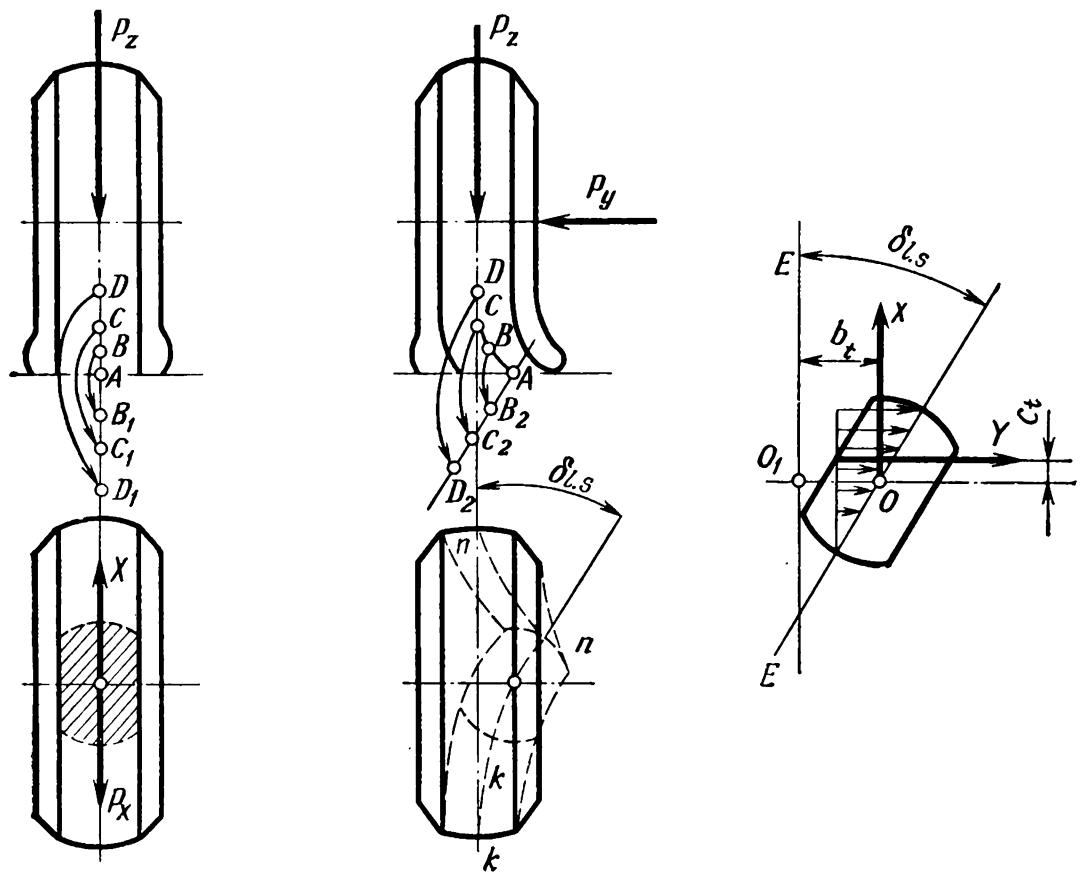


Fig. 14.1. Wheel lateral slip

wheel centre onto the road surface (point O_1) does not coincide with the centre of contact O, and the forces X and Y build up moments which tend to turn the wheel about the king pin. The value of these moments for the motion of a motor vehicle will be explained below.

Lateral slip can likewise be caused by setting the steerable wheels at an angle to the direction of motion (toe-in) and inclined vertically (camber).

When the values of P_y are low the sliding at the point of contact of the tyre with the road is negligible, and the wheel changes the direction of its motion mainly due to elastic deformations of the tyre. If the force P_y approaches in magnitude the grip force intensive sliding begins at the back of the contact area loaded most with transverse forces. When the force P_y is equal to the

grip force the tyre is likely to slide in the area of its contact with the road, which causes the wheel to move laterally. In this case, it is not lateral slip but an intricate wheel motion (rolling with sideway sliding) that takes place at a certain angle equal to the angle between the medium plane of the wheel and the direction of its motion.

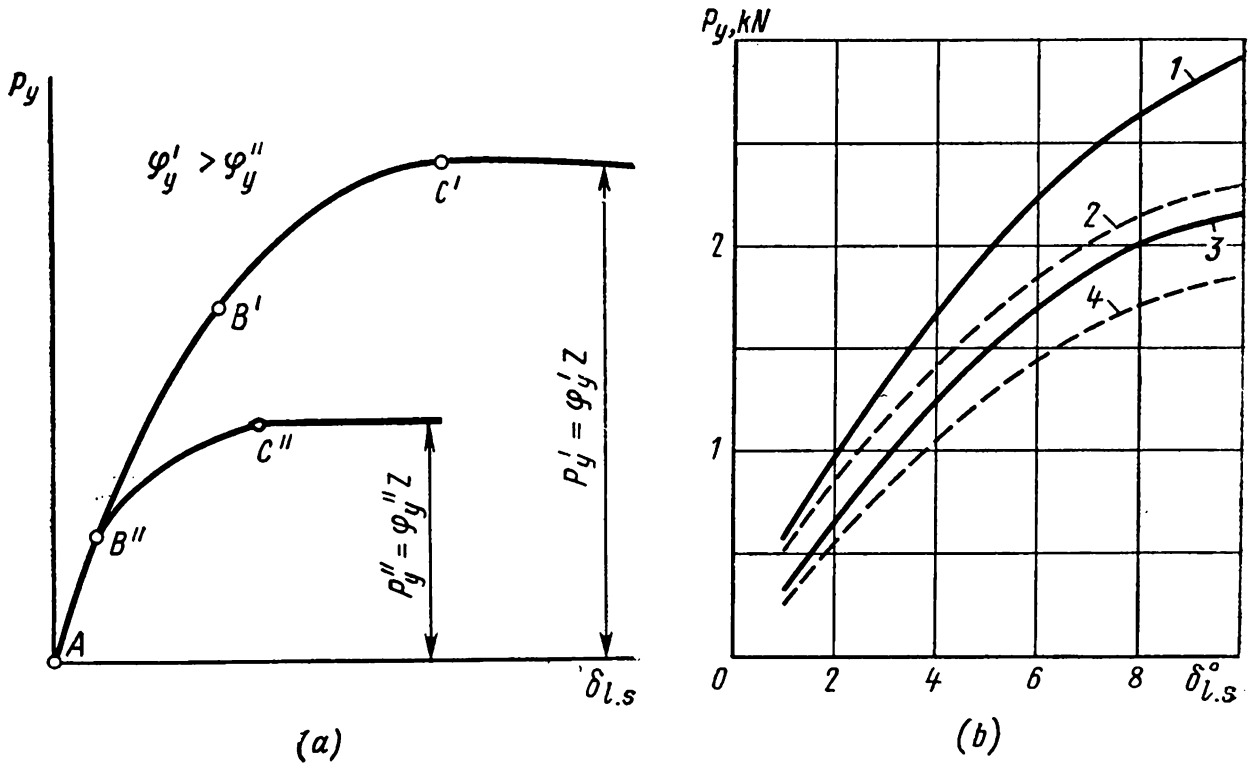


Fig. 14.2. Slip angle versus transverse force
(*a*) general view; (*b*) for a tyre 6.40-13 in size; *1* and *2*—load on the tyre equal to 6.0 kN; *3* and *4*—load equal to 2.0 kN; solid curves—pressure of 0.17 MN/m²; dash curves — pressure of 0.12 MN/m²

Figure 14.2*a* shows a tentative dependence diagram of the slip angle $\delta_{l.s}$ on the transverse force P_y applied to the wheel. When the values of the force P_y are small, the slip angle is approximately proportional to the force value and the section AB' may be regarded as linear. An increase in the P_y force causes sliding at the point of contact and the linear dependence is disturbed (section $B'C'$). If the force P_y grows to the value of the gripping force $\varphi_y Z$ full sliding of the tyre commences.

The dependence between the force P_y and the angle $\delta_{l.s}$ may be written as

$$P_y = k_{l.s}\delta_{l.s}$$

where $k_{l.s}$ is the *coefficient of resistance to lateral slip* showing what transverse force should be applied to the wheel for it to roll with a slip angle equal to 1 rad, in N/rad.

For small slip angles (up to 4-6°), the coefficient $k_{l.s}$ may be assumed approximately constant. For the tyres of cars, this coef-

ficient lies within 15-40 kN/rad and for the tyres of trucks and buses, within 30-90 kN/rad.

The less the coefficient of grip φ_y, the smaller the value of the transverse force causing full sliding of the wheel. Thus, the value of φ_y is about twice as large for the curve $AB'C'$ than for the curve $AB''C''$.

Figure 14.2*b* illustrates the experimental dependences of the slip angle on the transverse force for a tyre 6.40-13 in size with two values of the vertical load it takes: 2.0 kN (curves *3* and *4*) and 6.0 kN (curves *1* and *2*) with two values of air pressure in the tyre (solid curves correspond to a pressure of 0.17 MN/m^2 and dash curves, to a pressure of 0.12 MN/m^2). As can be seen from Fig. 14.2*b*, the dependences $P_y = f(\delta_{l.s})$ actually have a more complicated shape than the curves shown in Fig. 14.2*a*. For one thing, they have no straight section at the curve origin, and the coefficient $k_{l.s}$ may be taken as constant only with a certain approximation. An increase in the vertical load and tyre air pressure is attended by a growth in the resistance to the lateral slip of the tyre and the transverse force causing the full sliding.

In the case of lateral slip, the tyre is deformed not only in the radial but also in the cross direction, this increases the internal friction in the tyre. Besides, when the slip angles are large the particles of the tread intensively slip on the road. As a result, the force necessary for the wheel to roll during lateral slip sharply increases. According to experimental data, the force needed for wheel rolling with a lateral slip of 6-7° may be several times higher than the force that has to be applied to this wheel for rolling without lateral slip.

Lateral slip is investigated on stands or dynamometer bogies similar in design to those utilized to determine the coefficients f and φ. The tyre is adjusted at various angles to the direction of motion and the effect of speed, vertical load and other factors on lateral slip is determined.

During road tests, the slip angles are estimated when the motor vehicle moves in a straight path or over a circle. In the former case, a photocamera with its lens downwards is secured on the wheel axle so that the longitudinal boundaries of the frame are parallel to the longitudinal axis of the motor vehicle. As the vehicle moves, the road surface is photographed with a certain exposure and, as a result, separate spots and scratches on the road surface appear on the picture as lines. When the wheel rolls with a lateral slip (for example, due to crosswise inclination of the road) the lines on the picture are arranged relative to the frame boundaries at a certain angle equal to the tyre slip angle.

To find the slip angles of the axles during curvilinear motion, a tank with two pipes is mounted on the motor vehicle. The ends

of the pipes are secured in the middle of the vehicle axles close to the road. The tank is then filled with coloured water and air is pumped in to a pressure of 0.2-0.3 MN/m². When the motor vehicle circles the cock on the tank is opened and the water flows out from the pipes under pressure forming two circles on the road. The circumference of radius R_1 (Fig. 14.3) shows the path of motion of the front axle centre, and the circumference of radius R, the path of motion of the rear axle centre. If the speed is low the

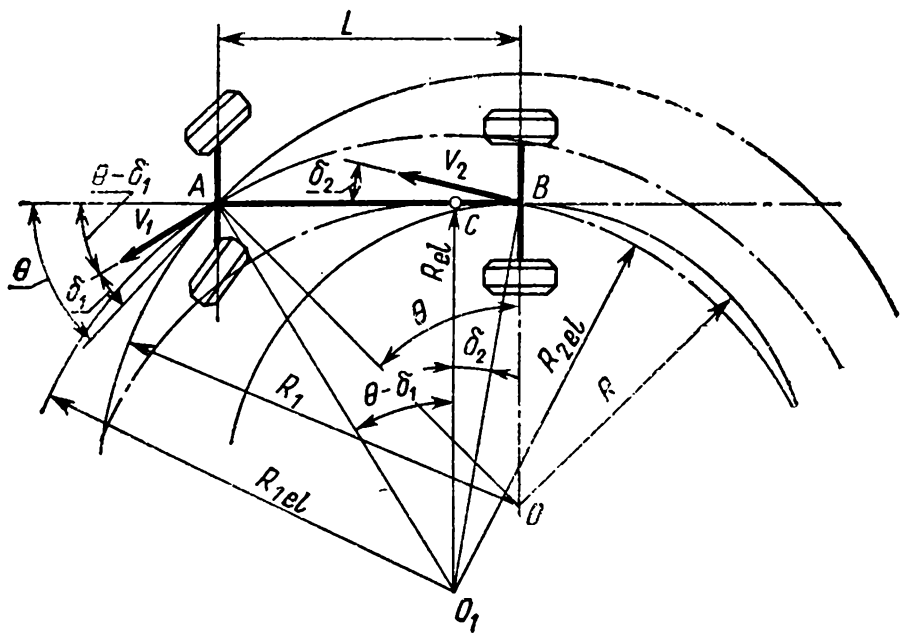


Fig. 14.3. Motion of motor vehicles with stiff and elastic tyres

lateral slip effect may be ignored, and it may be assumed that the vehicle runs over the same path on which it would move with tyres stiff in the cross direction. After measuring the radii R_1 and R_2, it is common practice to find the value of the mean angle θ showing the turn of the steerable wheels according to the formula:

$$\sin\theta = \frac{L}{R_1} = \frac{\sqrt{R_1^2 - R^2}}{R_1} \qquad (14.5)$$

Then, the test is repeated with the steerable wheels in the same position but at a higher speed. Due to lateral slip, the radii of the circumferences will become equal to R_{1el} and R_{2el} as shown in Fig. 14.3, and the centre of the circumference will shift from the point O to the point O_1. It follows from the right-angled triangles O_1CA and O_1CB that

$$R_{1el}\cos(\theta - \delta_1) = R_{2\,el}\cos\delta_2;$$

$$R_{1\,el}^2[1 - \sin^2(\theta - \delta_2)] = R_{2\,el}^2(1 - \sin^2\delta_2);\ R_{2\,el}\sin\delta_2 =$$

$$= L - R_{1\,el}\sin(\theta - \delta_1)$$

Squaring the latter equation and solving it together with the previous one, we obtain

$$\sin(\theta - \delta_1) = \frac{L^2 + R_{1\,el}^2 - R_{2\,el}^2}{2LR_{1el}} \tag{14.6}$$

$$\sin \delta_2 = \frac{L^2 + R_{2\,el}^2 - R_{1\,el}^2}{2LR_{2\,el}} \tag{14.7}$$

Having found the angle θ from formula (14.5), the slip angles can be determined from the two latter expressions.

Let us compare the steerability of motor vehicles with stiff tyres and tyres which are elastic in the transverse direction. For this purpose, we calculate the radius of turn of the middle of the vehicle rear axle with elastic and stiff tyres. In conformity with Fig. 14.3, it may be written that the base (in m)

$$L = R_{el} \tan(\theta - \delta_1) + R_{el} \tan \delta_2 \tag{14.8}$$

Since the angles θ, δ_1 and δ_2 are small, the radius (in m)

$$R_{el} = \frac{L}{\tan(\theta - \delta_1) + \tan \delta_2} \approx \frac{L}{\theta + \delta_2 - \delta_1} \tag{14.9}$$

For a motor vehicle with stiff tyres $\delta_1 = \delta_2 = 0$ and

$$R = \frac{L}{\tan \theta} \approx \frac{L}{\theta} \tag{14.10}$$

Therefore, if the path of motion of a motor vehicle with stiff tyres depends only on the angle of turn of the steerable wheels, then in a vehicle with transversely elastic tyres it also depends on the slip angles δ_1 and δ_2 A vehicle with elastic tyres can move curvilinearly under the action of the transverse force even if the steerable wheels are in their neutral position and the angle θ is equal to zero.

The turnability (ability to turn) is the property of a motor vehicle with elastic tyres to deviate, as a result of lateral slip, from the direction of motion determined by the position of the steerable wheels.

If the slip angles of the front and rear axles are mutually equal the turnability of a motor vehicle is known as *neutral*. Although, in this case, the radius $R = R_{el}$ according to formula (14.9), the path of motion of a motor vehicle with stiff tyres does not coincide with the path of motion of a vehicle having a neutral turnability because the centres of turn occupy different positions. By way of example, Fig. 14.4*a* illustrates the paths of motion of the gravity centre of motor vehicles with stiff (around centre O) and elastic tyres (around centre O_1). When a transverse force is acting on a vehicle with stiff tyres it preserves the former direction of mo-

tion if this force does not increase to the magnitude of the gripping force. A motor vehicle having a neutral turnability also moves straight under the action of the transverse force but at an angle of $\delta_{l.s}$ to the former direction of motion AA (Fig. 14.4*b*), i.e., along the line BB. To move in the direction AA, the driver should turn the vehicle so that its longitudinal axis is located at an angle to the direction of motion AA equal to the angle $\delta_{l.s}$.

If the angle $\delta_1 > \delta_2$ the radius $R_{el} > R$, and for a vehicle with elastic tyres to move over a curve of radius R the steerable wheels

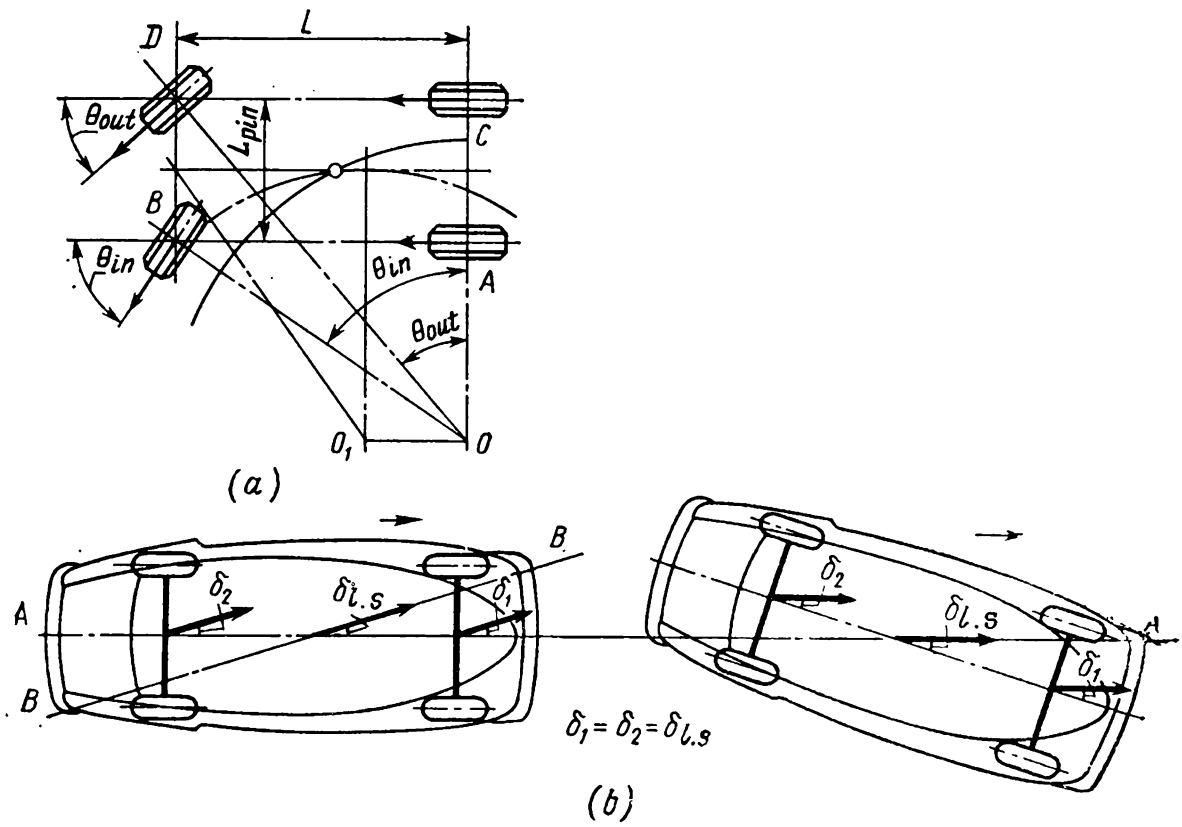


Fig. 14.4. Motion of a motor vehicle with neutral turnability
(*a*) curvilinear; (*b*) straight

should be turned through an angle larger than in the case of stiff tyres. In this case the turnability of a motor vehicle is called *inadequate*. In straight motion, the front axle of the vehicle with inadequate turnability begins to move under the action of the transverse force P_y (Fig. 14.5*a*) at an angle of δ_1 to the former direction of motion, and the rear axle at an angle of δ_2. Hence, a motor vehicle moves curvilinearly with the steerable wheels in the neutral position and turns around centre O_1. The turn generates a centrifugal force whose transverse component P_{cy} is now directed opposite to the force P_y, which diminishes the resultant transverse force and the lateral slip of the tyres. Therefore, a motor vehicle with inadequate turnability steadily preserves a straight direction of motion.

If the angle $\delta_1 < \delta_2$ and the radius $R_{el} < R$, then for a vehicle with elastic tyres to move over a curve of radius R the steerable wheels should be turned through an angle smaller than for a vehicle with stiff tyres. In this case, the turnability is called *excessive*. If a transverse force is applied to a motor vehicle with excessive turnability, in straight motion it will also take a curved

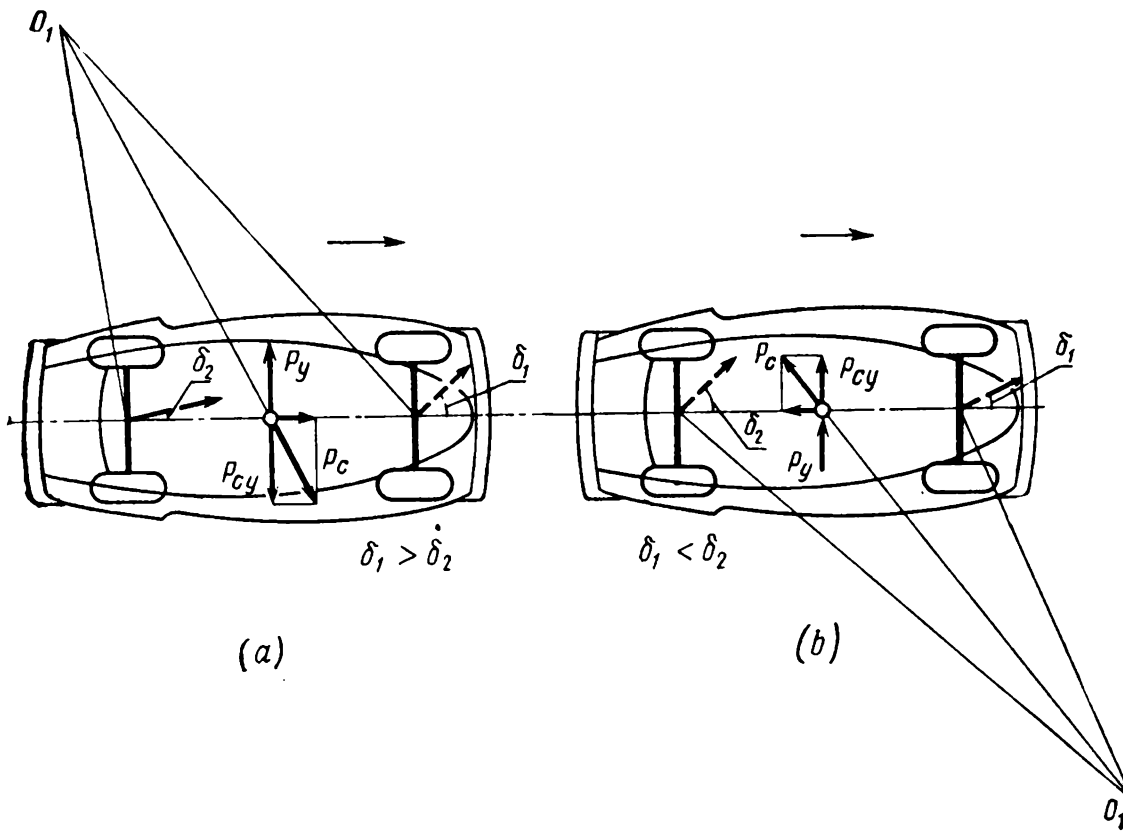


Fig. 14.5. Diagrams of motion of motor vehicles
(*a*) with inadequate turnability; (*b*) with excessive turnability

path (Fig. 14.5*b*). However, the transverse component P_{cy} of the centrifugal force lies, in this case, in the same direction as the transverse force P_y. This tends to increase the lateral slip and, therefore, the curvature of the path and the force P_{cy}, etc. If the driver fails to turn the steerable wheels in the required direction the centrifugal force may grow so much that the motor vehicle loses steerability or stability.

Thus, a vehicle with inadequate turnability is more stable and preserves the direction of motion better than a vehicle with excessive turnability. Besides, a motor vehicle with excessive turnability is likely to get out of control.

To explain this, let us determine the angle of turn (in rad) of the steerable wheels from formula (14.9):

$$\theta = \frac{L}{R_{el}} - (\delta_2 - \delta_1) \qquad (14.11)$$

The slip angles δ_1 and δ_2 are proportional to the transverse forces P_{y1} and P_{y2} which, in turn, are proportional to the square of speed v:

$$\delta_1 = \frac{P_{y1}}{k_{l.s1}} = \frac{M_1 v^2}{R_{el} k_{l.s1}} \tag{14.12}$$

$$\delta_2 = \frac{P_{y2}}{k_{l.s2}} = \frac{M_2 v^2}{R_{el} k_{l.s2}} \tag{14.13}$$

where $k_{l.s1}$, $k_{l.s2}$ are the coefficients of resistance to lateral slip of the front and rear axles, respectively, in kN/rad.

At a higher speed, the slip angles also increase, the angle δ_2 growing faster than the angle δ_1. This reduces the right side of expression (14.11) which becomes equal to zero at a certain, so-called, *critical speed* $v_{l.s}$. At this speed the motor vehicle takes a curvilinear path, although its steerable wheels are in the neutral position. If the speed exceeds $v_{l.s}$ the difference $(\delta_2 - \delta_1) > L/R_{el}$, and the angle θ becomes negative. This means that the front wheels should be steered leftwards to turn the motor vehicle rightwards. Hence, a motor vehicle with excessive turnability will lose control if its speed exceeds the critical speed $v_{l.s}$.

In order to determine this speed, we equate the angle θ to zero and insert into formula (14.11) the difference of the angles δ_2 and δ_1 obtained from expressions (14.12) and (14.13):

$$\delta_2 - \delta_1 = \left(\frac{M_2}{k_{l.s2}} - \frac{M_1}{k_{l.s1}}\right)\frac{v^2}{R_{el}} = \frac{L}{R_{el}}$$

Hence, the critical speed (in m/s)

$$v_{l.s} = \sqrt{\frac{L}{\frac{M_2}{k_{l.s2}} - \frac{M_1}{k_{l.s1}}}} \tag{14.14}$$

There is no critical speed in a motor vehicle with inadequate or neutral turnability, because the radicand is negative when $\delta_1 > \delta_2$ and the speed $v_{l.s}$ has an imaginary value, while it is equal to infinity when $\delta_1 = \delta_2$.

Example. Find the critical speed $v_{l.s}$ of a car moving without passengers if the coefficients of resistance to lateral slip of its axles are: $k_{l.s1} = 50$ kN/rad and $k_{l.s2} = 60$ kN/rad, and the masses are: $M_1 = 890$ kg and $M_2 = 720$ kg:

$$v_{l.s} = \sqrt{\frac{L}{\frac{M_2}{k_{l.s2}} - \frac{M_1}{k_{l.s1}}}} = \sqrt{\frac{2.7}{\frac{720}{60{,}000} - \frac{890}{50{,}000}}} = \sqrt{-467}$$

Hence, with such values of the coefficients of resistance to lateral slip, the motor vehicle has no critical speed and possesses inadequate turnability.

To ensure inadequate turnability of a motor vehicle, the air pressure in the tyres of the front wheels is somewhat reduced as

compared to the pressure in the rear wheels, thereby diminishing the value of the coefficient $k_{l.\,sl}$. Besides, the centre of gravity of the vehicle is slightly displaced towards the front axle, which increases the component of the centrifugal force acting on the steerable wheels.

An appreciable effect on steerability is exerted by the body lateral rolling which may cause the axis of the vehicle to turn

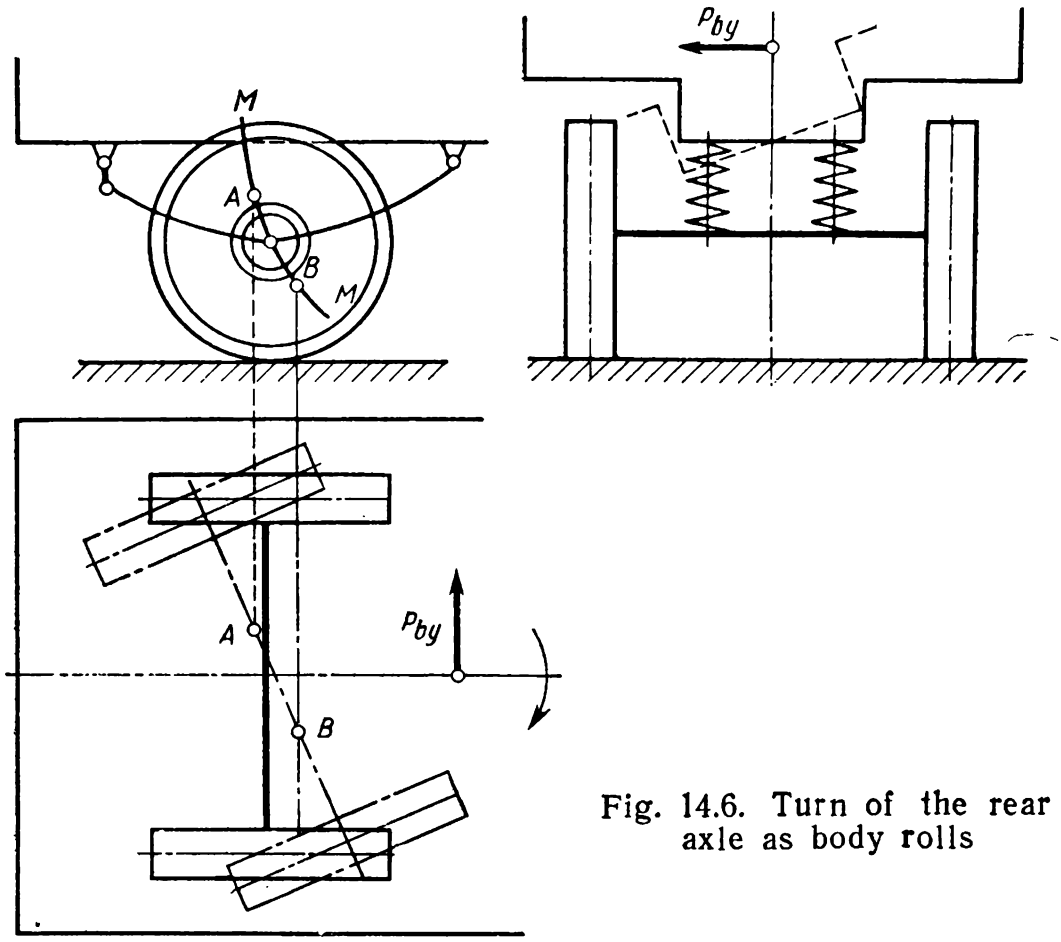


Fig. 14.6. Turn of the rear axle as body rolls

in a horizontal plane or change the camber of the steerable wheels.

Figure 14.6 shows a rear axle with a spring-loaded suspension of a vehicle making a right-hand turn. The front ends of the springs are attached to the body by an ordinary hinge and the rear ends, by a shackle. As the spring is deflected, the rear axle moves over the arc *MM*, its axis of swing being arranged near the hinge. The transverse force P_{by} inclines the body compressing the springs on one side of the vehicle (left-hand in Fig. 14.6) and straightening them on the other. The left-hand spring being compressed shifts the rear axle backwards (to point *A*) and the straightened-out right-hand spring moves this axle forward (to point *B*). As a result, the rear axle turns in a horizontal plane as shown by the dash line. This decreases the radius of turn and is tantamount to increasing the tendency of the vehicle to exces-

sive turnability. If the front end of the spring is connected to the body by a shackle and the rear end by a simple hinge it is possible to increase the inadequate turnability of the vehicle.

The effect produced by the camber of the steerable wheels on the steerability can be attributed to the lateral slip phenomenon. The slip angle due to camber is directly proportional to the camber angle α_{cam}. One degree of camber induces lateral slip equal on the average to an angle of 10-15′.

At the body lateral rolling of a vehicle with a spring-loaded suspension, the camber of the wheels hardly changes, and lateral slip arises only due to the action of the transverse force. In a motor vehicle with independent suspension, the wheel camber may vary within broad limits. For example, in the case of suspension on two cross arms (see Fig. 13.7) the body lateral rolling inclines the steerable wheels in the direction of action of the transverse force and thereby increases the lateral slip. The required turnability of a motor vehicle can be ensured using this specific feature of the suspension design.

14.4. Relation Between Angles of Turn of Steerable Wheels

When a motor vehicle is turned the speed vectors of the centres of all wheels are perpendicular to the radii drawn from the centre of turn O (see Fig. 14.4*a*). In a vehicle fitted with tyres stiff in the transverse direction, the speed vectors coincide with the medium planes of the wheels and the centre of turn is therefore located on the extension of the rear axle. The ratio between the angles θ_{out} of the outer wheel turn and θ_{in} of the inner wheel turn can be found from the triangles OAB and OCD:

$$\cot \theta_{out} = \frac{OC}{L}; \quad \cot \theta_{in} = \frac{OA}{L} \tag{14.15}$$

Subtracting the second equality from the first one, we get

$$\cot \theta_{out} - \cot \theta_{in} = \frac{(OC - OA)}{L} = \frac{L_{pin}}{L} \tag{14.16}$$

where L_{pin} is the distance between the centres of the king pins approximately equal to the track of the steerable wheels, in m.

It follows from equality (14.16) that the angles θ_{out} and θ_{in} are not equal. A wheel internal with respect to the centre of turn should be rotated through a larger angle than the outer wheel.

The steerable wheels are turned through various angles by a steering gear which cannot, however, satisfy accurately condition (14.16). For this reason, the steering gear is designed to satisfy equality (14.16) with rather large turn angles most frequently

met in operation. At the turn angles up to 12-15° both wheels turn approximately by the same angles. This allows the mean angle 0 to be used in calculations instead of the angles θ_{in} and θ_{out}, which is equal to half the sum of the latter angles, as has been done above.

If a motor vehicle is fitted with tyres which are elastic in the transverse direction, the speed vectors of the wheel centres do not coincide with the medium planes of the wheels but form slip angles with them. The centre of turn O_1 which is located as before at the intersection point of the perpendiculars to the speed vectors does not coincide with the centre of turn O of the motor vehicle fitted with stiff tyres. The ratio of the turn angles of the steerable wheels, which is necessary for rolling without lateral skidding, likewise changes. In this case, expression (14.16) will take the following form:

$$\cot\theta_{out} - \cot\theta_{in} = \frac{L_{pin}\left(L - \frac{M_2}{k_{l.s2}} v^2\right)}{\left[L - \left(\frac{M_2}{k_{l.s2}} - \frac{M_1}{k_{l.e1}}\right) v^2\right]^2} \qquad (14.17)$$

Formula (14.17) testifies to the fact that the ratio of the angles θ_{in} and θ_{out} in a vehicle with elastic tyres should be other than in a vehicle with stiff tyres.

It follows from the comparison of formulas (14.16) and (14.17) that in a vehicle with elastic tyres the angle θ_{out} differs from the angle θ_{in} less than in a vehicle with stiff tyres. This means that the form of the steering gear in a vehicle with elastic tyres should be close to a rectangle. Besides, a higher speed of motion should also change the ratio between the turn angles of the steerable wheels. This can be attained only by automatic steering linkage adjustment which by itself essentially complicates the design.

14.5. Oscillations of the Steerable Wheels

As a motor vehicle moves, the steerable wheels with the front axle can oscillate in a vertical plane and with the steering gear, around the king pins in a horizontal plane.

Angular oscillations (wobble) of the steerable wheels are not permitted because parts of the running gear and steering gear in this case take considerable alternating dynamic loads, and high amplitude oscillations result in a loss of control. Steady oscillations of the wheels, i.e., those continually repeated (self-excited), are especially dangerous. The mechanism of self-excitation is rather complicated and only a general idea will be given below.

When transverse angular oscillations (misalignment) of the front axle occur the steerable wheels incline and change their position relative to the vertical plane. A change in the position of the rotation axis of the wheels causes a gyroscopic moment M'_g which turns the steerable wheels around the king pins in a horizontal plane thus compressing the springs in the steering linkage joints. When the wheels are turned another gyroscopic moment M''_g appears which acts in a vertical plane and tends to increase the axle misalignment and inclination of the wheels.

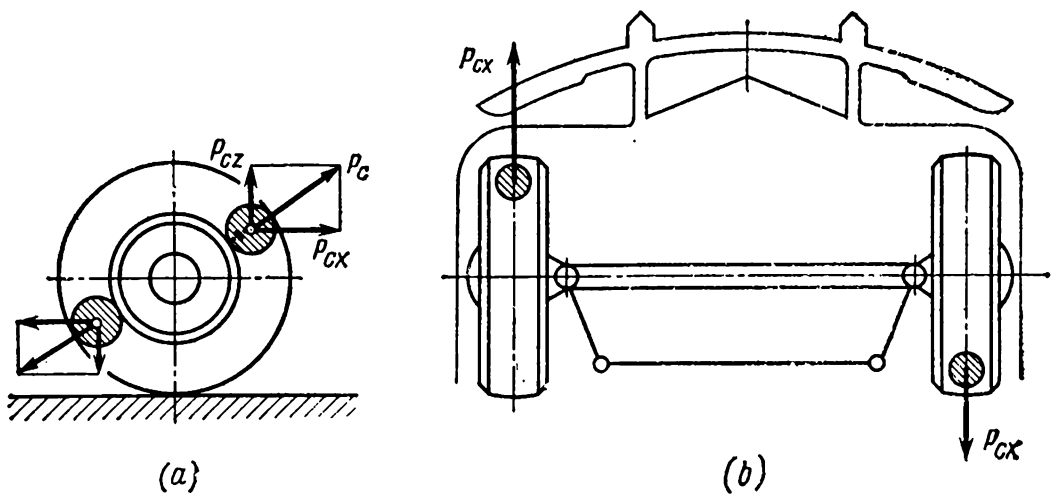


Fig. 14.7. Disbalance of the steerable wheels
(*a*) forces acting on unbalanced wheels; (*b*) appearance of a turning moment

Thus, the front axle misalignment causes angular oscillations (wobble) of the wheels which, in turn, increase the axle misalignment. Since both oscillating systems are interlinked, the oscillations of the front axle and the steerable wheels are strictly coordinated: if, for example, the left-hand wheel moves up it turns rightwards at the same time, if it goes down it turns to the left. Conversely, the right-hand wheel moving up turns to the left and in downward motion, rightwards. Hence, if the left-hand wheel tends to jump during misalignment the right-hand wheel is pressed stronger to the road and both wheels turn to the right.

The use of spring-loaded front suspensions, when the vertical motion of the steerable wheels does not cause them to incline excessively, is the principal way of reducing the gyroscopic moments and the resulting dangerous angular oscillations of the wheels.

These are intrinsic oscillations and depend only on the parameters of the system (characteristics of the elastic elements, moments of inertia and masses of separate parts). When a vehicle moves, the intrinsic oscillations may also be attended by forced oscillations caused by a periodically acting disturbing force. Such a force may, for example, appear due to disbalance (non-equilib-

rium) of the wheels. Rotation of an unbalanced wheel produces a centrifugal force P_c (Fig. 14.7*a*) which can be resolved into the vertical force P_{cz} and the horizontal force P_{cx}. The force P_{cx} tends to turn the wheel relative to the king pin and the force P_{cz} shift it in a vertical direction. Since the directions of the forces P_{cx} and P_{cz} are changed when the wheel rolls, the wheel begins to wobble. The most unfavourable case is when both wheels are unbalanced and the unbalanced portions are arranged in one plane but on different sides from the wheel axis of rotation (Fig. 14.7*b*). In this case, the turning moments acting on the wheels add up and the angular oscillations appreciably grow in magnitude.

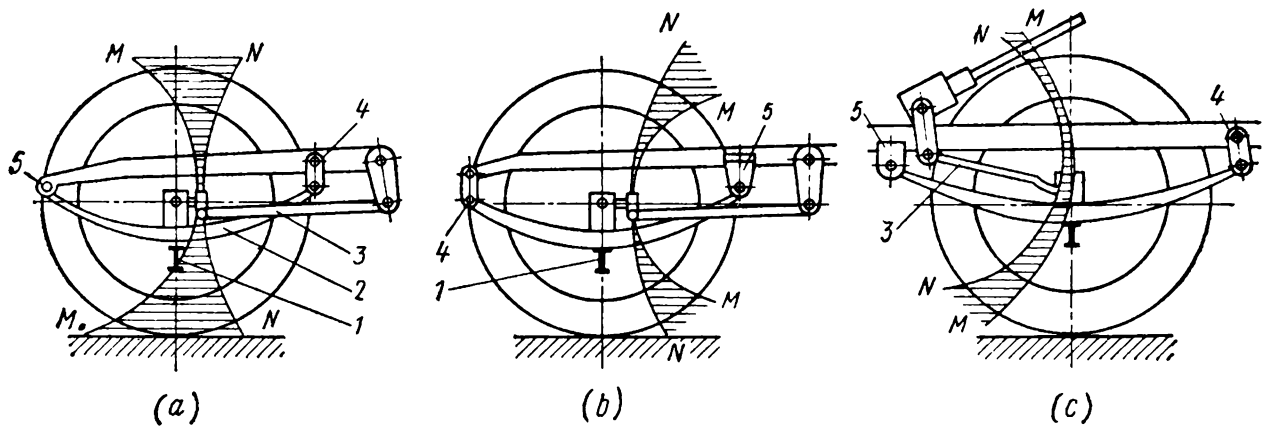


Fig. 14.8. Oscillations of wheels when the rocking centres of the front axle and the drag rod do not coincide

(*a*) spring shackle located behind; (*b*) spring shackle located in front; (*c*) steering gear before front axle; *1*—front axle; *2*—spring; *3*—steering rod; *4*—shackle; *5*—ordinary hinge

Disturbing forces arise when a wheel runs onto irregularities which can appear at approximately equal path intervals. If a motor vehicle moves on a road with such a surface the frequencies of the forced and intrinsic oscillations at a definite speed will be identical, i.e., a resonance will set in at which the oscillations will continually grow and the driver has to drop the speed to keep the vehicle under control.

Wobble of the steerable wheels may also be caused by the dual linking of these wheels to the motor vehicle body (with the aid of the steering linkage and suspension). When spring *2* (Fig. 14.8*a*) connected with the frame by a simple hinge *5* in front and by shackle *4* in the rear bows, the front axle *1* moves over the arc *MM*, its rocking axis is near hinge *5*. The front end of drag rod *3* which rocks relative to the steering arm pin describes arc *NN*. The arcs *MM* and *NN* diverge and for this reason the wheels move in a vertical plane and turn about the king pins, thus impairing the vehicle control and tiring the driver. To lessen wheel wobble, the paths of motion of the front axle and the front end

of the drag rod should be brought closer. With this aim in view, the front end of the spring is connected with the frame by a shackle, and the rear end, by a simple hinge (Fig. 14.8*b*), or the steering mechanism is located ahead of the front axle (Fig. 14.8*c*).

14.6. Stabilization of the Steerable Wheels

Stabilization of the steerable wheels is their property to preserve neutral position and automatically return to it.

It is difficult to steer a poorly stabilized motor vehicle, because it constantly tends to move away from a straight path and the driver has to continually correct the steering to maintain straight motion. All these corrections fatigue the driver and intensify wear of the tyres and the steering mechanism. In a well-stabilized motor vehicle the steerable wheels steadily preserve their neutral position and allow a straight run even if the driver does not steer. When coming out of a turn the steerable wheels of such a vehicle return to the neutral position without any effort on the part of the driver. This fact makes driving easier and reduces the hazard of collision with obstacles situated on paths with small radii of curvature.

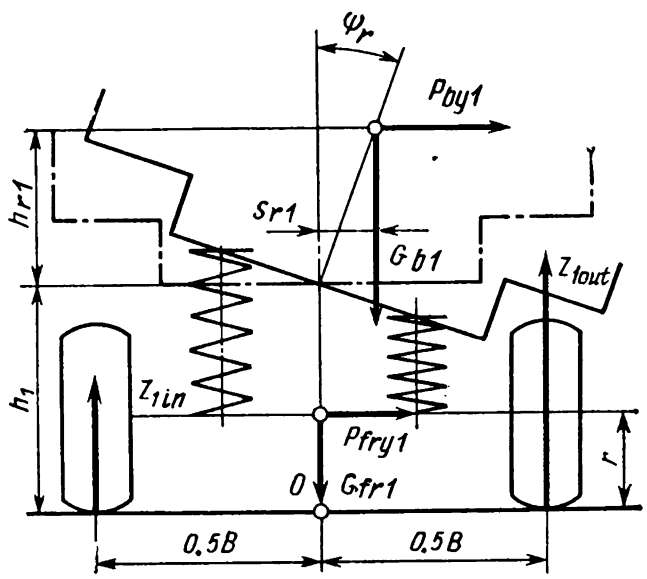


Fig. 14.9. Forces acting on the front part of a motor vehicle

The process of stabilizing the steerable wheels is extremely complicated. It is especially difficult to investigate the stabilization during a straight run, because it is altogether impossible to determine analytically the values and frequencies of the road roughness forces taken by the wheels. For this reason, only stabilization of the wheels when the motor vehicle comes out of a turn is considered below. The stabilization measure, in this case, is the angular velocity of the steering wheel as it returns to the neutral position. During tests, as the motor vehicle runs uniformly over a circumferential arc, the driver lets off the steering wheel and it returns together with the steerable wheels to the neutral position. The change in the angle $\theta_{s.w}$ of the steering wheel turn in time is registered by a recorder. The greater the angular velocity of the steerable wheels returning to the neutral position, the better they are stabilized.

The speed of return of the steerable wheels depends on the stabilizing moment built up by the road reactions and caused by the longitudinal and cross inclination of the king pins and also by the transverse elasticity of the tyres.

Let us determine the road reactions acting on the steerable wheels. As a motor vehicle moving on a level road is turned, its front is acted upon by the centrifugal forces of the spring-loaded P_{by1} and spring-free P_{fry1} masses (Fig. 14.9).

We write an equation of moments of all forces relative to the point O:

$$(Z_{1out} - Z_{1in})\,0.5B = P_{fry1}r + P_{by1}(h_{r1} + h_1) + G_{b1}s_{r1} \quad (14.18)$$

Ignoring the small value of the moment $P_{fry1}r$ and taking into account formulas (13.20) and (13.21), we obtain

$$0.5(Z_{1out} - Z_{1in})B = c_{ang1}\psi_r + P_{by1}h_1 \quad (14.19)$$

where c_{ang1} is the angular rigidity of the front suspension in N·m/rad.

If the vehicle body does not deform the roll angles of its front and rear are mutually equal. Then, for the rear part

$$0.5(Z_{2out} - Z_{2in})B = c_{ang2}\psi_r + P_{by2}h_2 \quad (14.20)$$

where c_{ang2} is the angular rigidity of the rear suspension in N·m/rad.

The increments in normal reactions acting on each front and rear wheel (in N) are

$$\Delta Z_1 = 0.5(Z_{1out} - Z_{1in}) = \frac{c_{ang1}\psi_r + P_{by1}h_1}{B} \quad (14.21)$$

$$\Delta Z_2 = 0.5(Z_{2out} - Z_{2in}) = \frac{c_{ang2}\psi_r + P_{by2}h_2}{B} \quad (14.22)$$

The normal road reactions acting on the wheels

$$\left.\begin{aligned} Z_{1out} &= 0.5G_1 + \Delta Z_1;\ Z_{1in} = 0.5G_1 - \Delta Z_1 \\ Z_{2out} &= 0.5G_2 + \Delta Z_2;\ Z_{2in} = 0.5G_2 - \Delta Z_2 \end{aligned}\right\} \quad (14.23)$$

The normal reactions on corner are proportional to the transverse force P_{by} which is, in turn, proportional to the angle θ. For this reason, the normal reactions with small values of θ, i.e., in motion over curves with relatively large curvature radii may be regarded as linear functions of the angle θ.

In uniform motion, the tangential reaction X for the driven wheel is the force of resistance to rolling and is equal to the product fZ_1.

The sum of the road transverse reactions on both front wheels is equal to the force P_{y1}. If the effect of the vertical load is

neglected it may be assumed that the rigidities of tyres of both front wheels are identical in the transverse direction. The transverse deformations are also mutually equal because the wheels are rigidly linked by the front axle. Hence, the transverse forces and the road reactions taken by each of the front wheels are approximately equal and it may be assumed that the transverse force is shared equally by the front wheels:

$$Y_{1out} = Y_{1in} = \frac{P_{y1}}{2} = \frac{M_1 v^2}{2L}\theta \qquad (14.24)$$

where Y_{1out} and Y_{1in} are the road transverse reactions acting on the outer and inner front wheels, respectively.

The force P_{y1} is directly proportional to the angle θ as are, therefore, the forces Y_{1out} and Y_{1in}.

Example. Determine the road reactions acting on the front wheels of a vehicle if $P_y = 5000$ N; $P_{by} = 4500$ N; $\psi_r = 0.1$ rad; $f = 0.025$; $c_{amg1} = 18$ kN·m/rad; $h_1 = 0.1$ m; $b_b = 1.4$ m; $G_1 = 8500$ N; $b = 1.3$ m; $M_1 = 850$ kg; $B = 1.47$ m.

The transverse force acting on the front part of the body

$$P_{by\,1} = P_{by}\frac{b_b}{L} = 4500\,\frac{1.4}{2.7} = 2330 \text{ N}$$

The normal reactions on the front wheels

$$Z_{1out} = 0.5G_1 + \frac{c_{amg1}\psi_r + P_{by1}h_1}{B} =$$

$$= 0.5 \times 8500 + \frac{18{,}000 \times 0.1 + 2330 \times 0.1}{1.47} = 5630 \text{ N}$$

$$Z_{1in} = G_1 - L_{1out} = 8500 - 5630 = 2870 \text{ N}$$

The tangential reactions on the front wheels

$$X_{1out} = fZ_{1out} = 0.025 \times 5630 = 141 \text{ N}$$

$$X_{1in} = 0.025 \times 2870 \approx 72 \text{ N}$$

The transverse reactions on the front wheels

$$Y_{1out} = Y_{1in} = 0.5P_y\frac{b}{L} = 0.5 \times 5000\,\frac{1.3}{2.7} \approx 1200 \text{ N}$$

If the values of the reactions are known the stabilizing moment can also be found. Figure 14.10 shows the front inner wheel of a motor vehicle turned through angle θ from the neutral position (from the axis Oy). The king pin is inclined in the longitudinal direction at an angle of γ_t and in the transverse direction at an angle of β_t and forms an angle of μ_t with the vertical line, and

$$\tan \mu_t \approx \sqrt{\beta_t^2 + \gamma_t^2} \qquad (14.25)$$

The projection of the king pin axis on the road surface forms an angle of λ_t with the axis Oy. Since the angles β_t and γ_t are small, then

$$\sin\lambda_t \approx \frac{\gamma_t}{\mu_t}, \quad \cos\lambda_t \approx \frac{\beta_t}{\mu_t} \tag{14.26}$$

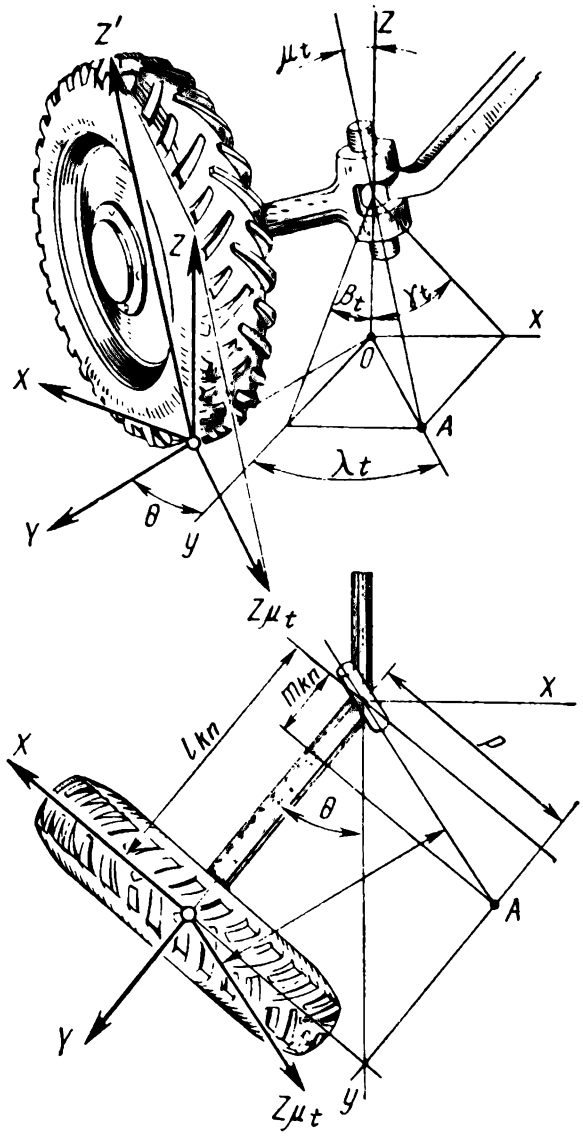


Fig. 14.10. Forces acting on steerable wheel

To determine the moment of a force relative to an axis, one should find the moment of this force with respect to any point on the axis and project the vector of the obtained moment onto the direction of this axis. But since the angles β_t and γ_t are small, it will be sufficient to estimate only the moment relative to point A where the axis of the king pin intersects the surface of the road.

According to Fig. 14.10, at an angle of camber $\alpha_{cam} = 0$ the moments of reactions X and Y relative to point A (in N·m) are

$$M_x = X(l_{kn} - m_{kn}) = \\ = X[l_{kn} - r\mu_t \cos(\theta + \lambda_t)] \tag{14.27}$$

$$M_y = YP = Yr\mu_t \sin(\theta + \lambda_t) \tag{14.28}$$

where l_{kn} is the length of the steering knuckle axis in m.

To determine the moment of reaction Z, let us resolve it into the force Z' which is parallel to the king pin and creates, therefore, no moment relative to its axis, and the force $Z\mu_t$ lying in the plane of the road and parallel to the line AO. Then,

$$M_z = Z\mu_t n = Z\mu_t l_{kn} \sin(\theta + \lambda_t) \tag{14.29}$$

Expressing the sine and cosine of the sum of the angles θ and λ_t through the sines and cosines of these angles and accounting for expressions (14.26), we obtain for the inner wheel

$$M_x = X[l_{kn} - r(\beta_t \cos\theta - \gamma_t \sin\theta)] \tag{14.30}$$

$$M_y = Yr(\beta_t \sin\theta + \gamma_t \cos\theta) \tag{14.31}$$

$$M_z = Zl_{kn}(\beta_t \sin\theta + \gamma_t \cos\theta) \tag{14.32}$$

When one wheel is turned in the positive direction (inside) the other wheel turns in the negative direction (outside). Since $\sin(-\theta) = -\sin\theta$, the signs of the terms containing $\sin\theta$ in the last formulas should be changed to determine the moments acting on the outer wheel. Thus, both steerable wheels are acted upon by six moments of the road reaction (in N·m). The inner wheel is acted upon by the total moment

$$M_{in} = M_{x\,in} + M_{y\,in} + M_{z\,in} \tag{14.33}$$

and the outer wheel, by the total moment

$$M_{out} = M_{x\,out} + M_{y\,out} + M_{z\,out}$$

Figure 14.11 shows the forces and moments acting on the steerable wheels of a motor vehicle. The points of intersection of

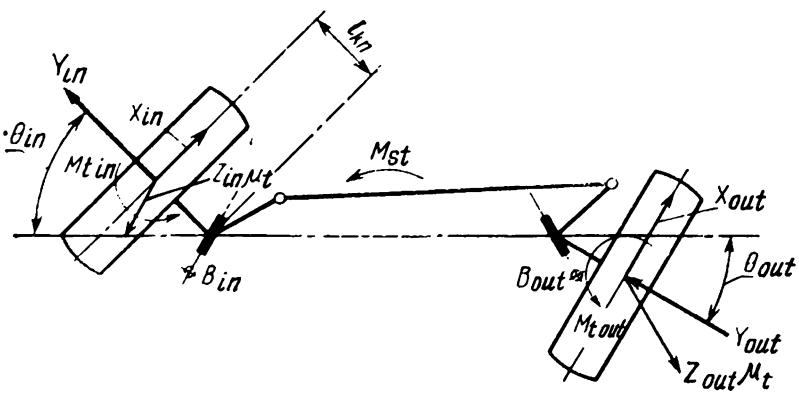


Fig. 14.11. Forces and moments acting on steerable wheels

the axes of the king pins with the road surface are denoted by the letters B_{in} and B_{out}. The moments of the tangential reactions of the road X_{in} and X_{out} with respect to these points are directed oppositely and are partly mutually equalized as are the moments of the components $Z_{in}\mu_t$ and $Z_{out}\mu_t$ of the vertical reactions. Therefore, the moments of these forces exert a negligible effect on stabilization. The moments of the transverse reactions of the road Y_{in} and Y_{out} go in one direction and, on being summed, produce a noticeable effect on the stabilization of the steerable wheels.

When a wheel rolls with lateral slip its tyre is also acted upon by the moments M'_x and M'_y equal to Xb_t and Yc_t (see Fig. 14.1). The sum of the moments $M'_x + M'_y = M_t$ is called the *tyre stabilizing moment.*

Figure 14.12*a* illustrates the experimentally found dependences of the moment M_t on the wheel slip angle for some models of Soviet-made tyres. As the slip angle grows, the stabilizing moment of the tyre first increases, and then diminishes after reaching its maximum value. This course of the curves can be attributed to the fact that an increase in the slip angle is caused, on the one hand, by a higher transverse force Y and its moment Yc_t and a higher force of resistance to rolling and a transverse shift of the

tyre, on the other. The moments Xb_t and Yc_t have different directions and this reduces the moment M_t at large slip angles.

Summing up all the moments acting on the inner and outer steerable wheels and taking into account their directions, we obtain the stabilizing moment M_{st} applied to the steering gear arm.

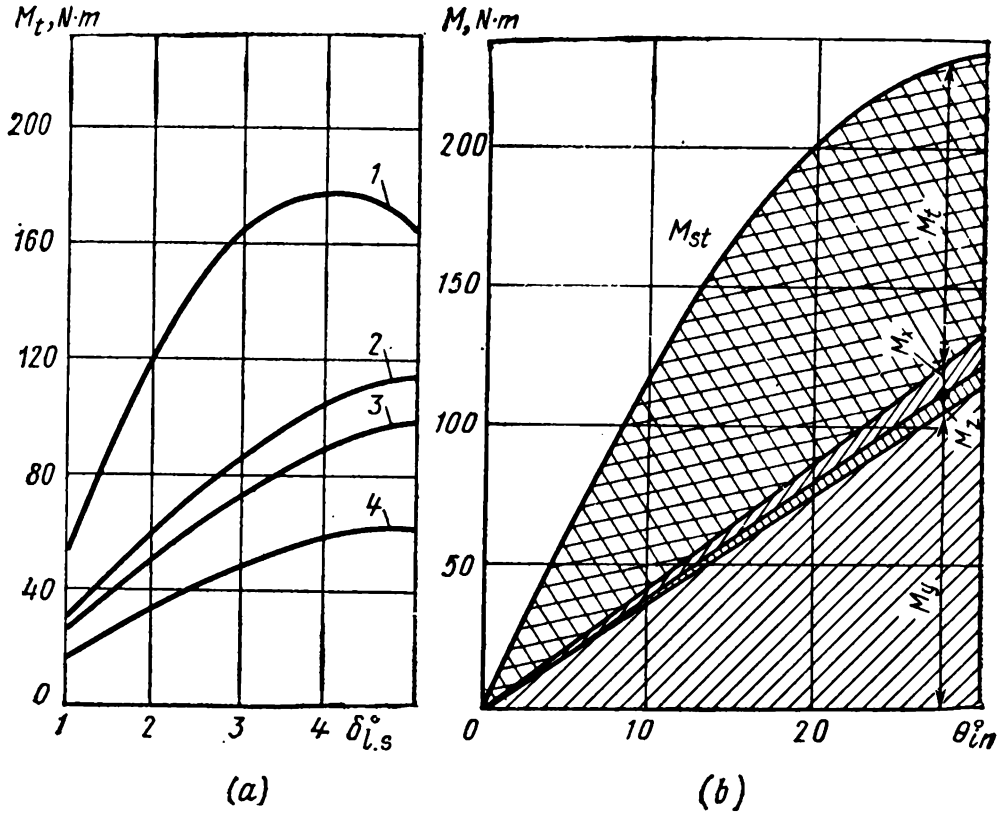


Fig. 14.12. Stabilizing moment
(*a*) with tyres of various sizes; (*b*) of a vehicle being designed; *1*—7.50-16; *2*—6.00-15; *3*—7.00-15; *4*—5.00-16

The steering arm is also acted upon by the moment $M_{s.c}$ of the forces of resistance in the steering, which opposes the motion of parts. When a motor vehicle begins a turn the driver should create a moment of such magnitude on the steering wheel so as to overcome the total moment $M_{st} + M_{s.c}$. For this reason, to make the steering easier the moment M_{st} should not be too high.

When a motor vehicle comes out of a turn the driver lets off the steering wheel and the front wheels return to the neutral position under the action of the difference in the moments $M_{st} - M_{s.c}$. Hence, friction in the steering spoils stabilization.

During a straight run, the stabilizing moments on the right- and left-hand steerable wheels are mutually equalized and the stabilizing moment on the steering gear arm is equal to zero. In this case, the wheels are mainly stabilized by the moment $M_{s.c}$ which does not allow them to deflect from the neutral position.

Figure 14.12*b* shows the change in the moment M_{st} and its components depending on the angle of turn of the inner steerable

wheel ($\beta_t = 6°$, $\gamma_t = 3.5°$). With such ratio between the angles of inclination of the king pin, the moments M_t and M_y display the maximum value. The moment M_z is comparatively low and the moment M_x is so small that it affects stabilization of the wheels only when the vehicle moves on a road with a high f coefficient or during braking, i.e., when the tangential reactions are large.

Thus, stabilization of the steerable wheels is principally affected by the transverse elasticity of the tyre and the longitudinal inclination of the king pin whose change alters the moments M_t and M_y. The stabilizing moment practically depends on the transverse inclination of the king pin only with large values of angle θ, i.e., during motion along curves of small radii.

Steerability of a motor vehicle depends on the technical state of its running gear and steering. A drop of air pressure in the tyre of one of the steerable wheels increases its resistance to rolling and diminishes the transverse rigidity, thus causing the vehicle to deflect towards the tyre with the reduced pressure. Wear of the king pin joints and parts of the steering gear increases clearances in the kinematic links and causes arbitrary oscillations of the wheels. When clearances are large the wobble and jumps of the steerable wheels may grow so high that the grip of the wheels with the road may be violated. Oscillations may be caused by out-of-balance of the wheels, this frequently happens when tyres repaired by vulcanized patches are used. As a rule, the repaired spot has a larger mass as compared with the nearby tyre sections and makes the rolling wheel wobble.

Large clearances in wheel bearings and king pin joints impair stabilization of the steerable wheels both in straight motion and also when a motor vehicle comes out of a turn. Excessive clearances between the king pins and the bushings require more turns of the steering wheel and increase their amplitudes.

Steerability of cars may be impaired by residual deformation of the front suspension springs. Their sag shifts the suspension arms and changes the wheel camber angles and transverse tilt of the king pins. If one of the springs sags the angles β_t and α_{cam} change only on one side of the car. In this case, the stabilizing moments on the right- and left-hand steerable wheels in a straight run are not mutually equalized and the car is entrained aside.

Stabilization of the steerable wheels may also be worsened in view of a wrongly regulated steering control. Excessive tightening of the springs in the joints of the drag rod, taper bearings of the worm and the couple of the steering mechanism tends to increase the friction moment $M_{s.c}$. This prevents the wheels from returning to the neutral position and makes the car control difficult.

Chapter XV CROSS-COUNTRY ABILITY OF A MOTOR VEHICLE

15.1. General on Cross-Country Ability

Motor vehicles should be sufficiently versatile to run not only on good roads with hard surfaces but also on dirt roads and across country.

The design of motor vehicles should conform to the conditions for which they are mainly intended. Thus, intercity buses and high-class cars are designed to operate on level roads with hard surfaces and minimum requirements are imposed on them as to their versatility. Trucks, most cars and buses with a small holding capacity are used not only in cities and rural localities on roads with hard and soft surfaces which may be either in a good or bad state but also where there are no roads at all. Trucks employed in agriculture, construction, and forestry operate in especially arduous conditions. For this reason, their cross-country ability should be much higher and represent one of their principal performances. On bad roads and across country the driver has to overcome various obstacles and additional resistances: steep ascents and descents, soft and loose soil, stones, stumps, pits, ditches, water barriers, slippery roads, etc.

Until now there is no criterion which would make it possible to estimate fully the vehicle cross-country ability. It has been found that the cross-country ability depends on many factors mainly describing the supporting and tractive properties and the geometrical parameters of the vehicle, and also the design of separate units in the transmission (differential gear, gearbox, etc.). Besides, cross-country ability is influenced, to a certain degree, by such performances as stability and manoeuvrability.

It should also be noted that the quality of driving has a noticeable effect. Obviously, all other conditions being equal, a skilled driver will make his vehicle more versatile.

According to their cross-country ability all motor vehicles are arbitrarily subdivided into three groups. The classification is based on the ratio between the total number of the wheels of a vehicle and the number of its driving wheels expressed by a "wheel formula". This formula consists of two figures, the first figure representing the total number of wheels and the second, the number of the driving wheels. For example, the wheel formula

for ЗИЛ-130 takes the form 4 × 2; for ГАЗ-66, 4 × 4; for ЗИЛ-131, 6 × 6, etc.

The first group includes vehicles with a limited (normal) cross-country ability, types 4 × 2 and 6 × 2; the second group, with improved cross-country ability, types 4 × 4, 6 × 4 and 6 × 6; and the third group, all-terrain vehicles, type 8 × 8, having a special layout.

15.2. Trafficability Test

Motor vehicles intended to be tested for cross-country ability (trafficability) should be technically in good repair. The tests are conducted under full load established for the given road conditions, the load being distributed so that the vertical loads taken by the axles are nominal. Before the tests, the units of the motor vehicle should be warmed up to the normal temperature. The units are warmed up as the vehicle runs on the road. During the test, the air pressure in the tyres is maintained in conformity with the accepted ratings. Motor vehicles equipped with a centralized system of type pressure control are tested at various pressures.

Tests are usually carried out on level sections not less than 50 m long. The humidity, strength, density, and thickness of the loosened layer of soil should be the same over the entire section. Before the test is begun, the characteristic of the soil is determined for each section. An expert on road-building should always be present when a motor vehicle is tested for cross-country ability.

Motor vehicles of limited cross-country ability (general-purpose vehicles) are tested on harrowed land with a ploughed depth of 15-20 cm, on moist sand, virgin snow 20-25 cm deep and on a solid dirt road.

Vehicles of improved cross-country ability are tested on harrowed land with a 20-25 cm ploughed layer and a 50-60 per cent moisture content, on dry quicksand with a loosened layer 30-40 cm deep, on boggy land with a carrying capacity (specific resistance of the soil to vertical load) of 0.09-0.11 MN/m^2, on granular loose virgin snow 30-40 cm deep and on a soaked dirt road with a 10-15 cm mud layer.

All-terrain motor vehicles are tested in still more arduous conditions: on strongly soaked plough-land with a ploughing depth of 25-30 cm and 70 per cent moisture content, on dry sand with a loosened layer 40-50 cm deep, on boggy land with a carrying capacity of 0.05-0.07 MN/m^2 and also on granular loose virgin snow 50-60 cm deep and on a strongly soaked dirt road with a 25-30 cm layer of slippery mud.

The tests are intended to determine the *geometrical, supporting and tractive characteristics.*

In order to establish the supporting and tractive characteristics, it is necessary to find the grip and rolling resistance coefficients and also the *specific tractive power*, i.e., the ratio of the maximum tractive force developed on the vehicle pulling hook just before the wheels begin to skid or the motor stalls to the total weight of the motor vehicle. These characteristics are measured with dynamometers when the vehicle is towed or by tensometers which

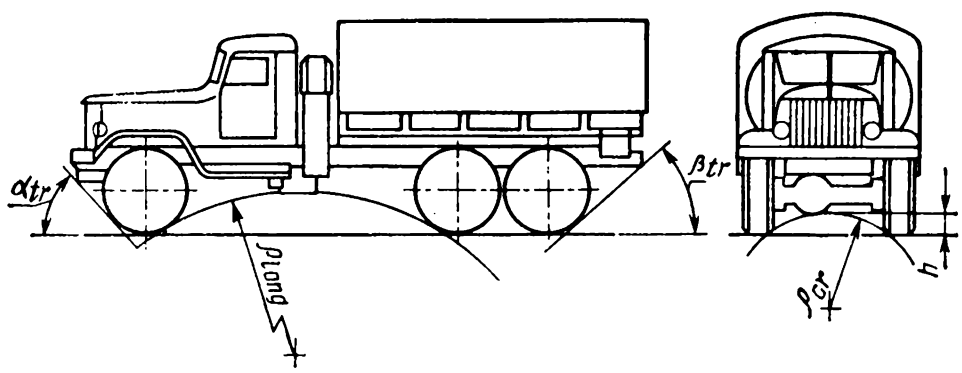


Fig. 15.1. Geometrical characteristics of cross-country ability

record the torques and reactions on the wheels of the motor vehicle being tested.

Motor vehicles provided with a centralized system of tyre pressure control are tested at a pressure which corresponds to the minimum rolling resistance losses.

To establish the characteristics of geometrical trafficability, one should find the *road clearances h* (Fig. 15.1), *the front* α_{tr} *and rear* β_{tr} *angles of trafficability, the radii of longitudinal* ρ_{long} *and cross-*ρ_{cr} *trafficability*, etc.

As a rule, a vehicle cross-country ability is estimated by observing its motion on bad roads. In this case, the number of stops due to various causes is recorded (because of insufficient tractive force, skidding, low parts knocking against obstacles, etc.) along with the speed and fuel consumption under various loads.

Cross-country ability can be estimated comparatively by means of the *trafficability index* η_{tr} which is the ratio of the maximum efficiency of the vehicle determined experimentally when it moves on a bad road to its efficiency on a road with a good (asphalt) surface:

$$\eta_{tr} = \frac{G_{bad} v_{bad}}{G_{good} v_{good}}$$

where G_{bad}, v_{bad} = actual load and mean speed of the vehicle in bad road conditions (in a given locality)

G_{good}, v_{good} = nominal load and mean speed of the vehicle in good road conditions

15.3. Trafficability Requirements

The dynamic factor of motor vehicles should be high enough to overcome the increased resistance to motion. To increase the tractive force, all or most of the wheels should be driving and the vertical load they take distributed equally. Skidding of the wheels can be eliminated or restricted within the necessary limits that ensure the best soil grip by specially designed elements in the transmission and selection of a proper pattern of the tyre tread.

The pressure of the wheels on the surface should be such that the vehicle can traverse soil of low carrying capacity. As the motor vehicle moves, this pressure should be regulated as required. The road clearances and other geometrical characteristics should not hamper movement of the vehicle on uneven roads or overcoming concentrated obstacles.

For a motor vehicle to move along narrow thorough-fares and roads, its steerable wheels should turn through possibly larger angles. The suspension must be designed so that appreciable misalignment of the axles may occur when the vehicle moves on uneven roads. This ensures good adaptability of the wheels to terrain irregularities. The engine, transmission, and the units of the running gear and controls should be reliably protected against water, dirt and dust.

A motor vehicle should carry devices which improve its cross-country ability, such as a winch and antiskid chains.

Steering control should be as easy as possible to lessen the driver's fatigue especially on heavy roads.

15.4. Geometrical Characteristics of Trafficability

Clearance is the distance h between one of the lowermost points of the motor vehicle and the road surface (see Fig. 15.1) which characterizes the possibility of motion without knocking against stones, stumps, etc.

The front α_{tr} *and rear* β_{tr} *trafficability angles* describe the vehicle cross-country ability on uneven roads when it climbs or leaves an obstacle, for example, when traversing hillocks, ditches, etc. To determine the α_{tr} and β_{tr} angles, straight lines are drawn which are tangential to the external circumferences of the tyres of the front and rear wheels and to the farthest points on the front and rear portions of the vehicle.

The longitudinal ρ_{long} *and cross* ρ_{cr} *radii of trafficability* (see Fig. 15.1) determine the contours of an obstacle which can be overcome by a vehicle without touching it. The radii are evaluated on the sketch of the motor vehicle drawn to scale from the radii

of the corresponding circumferences drawn tangentially to the external circumferences of the tyres and the lowermost point of the vehicle within the limits of the base (ρ_{long}) or track (ρ_{cr}). The smaller the longitudinal and cross trafficability radii, the better the general traversing characteristics of the vehicle. For example, the radius ρ_{long} can be reduced by shortening, for example, the vehicle base. The values of the ρ_{long} radius (in m) for some motor vehicles are illustrated below.

Cars, type 4×2	3.2-8.3
Trucks, type 4×2	2.7-5.5
Trucks, types 4×4, 6×4, 6×6	1.9-3.6

Manoeuvrability is the property of a motor vehicle to turn within a minimum area. *The minimum radius of turn of the outer*

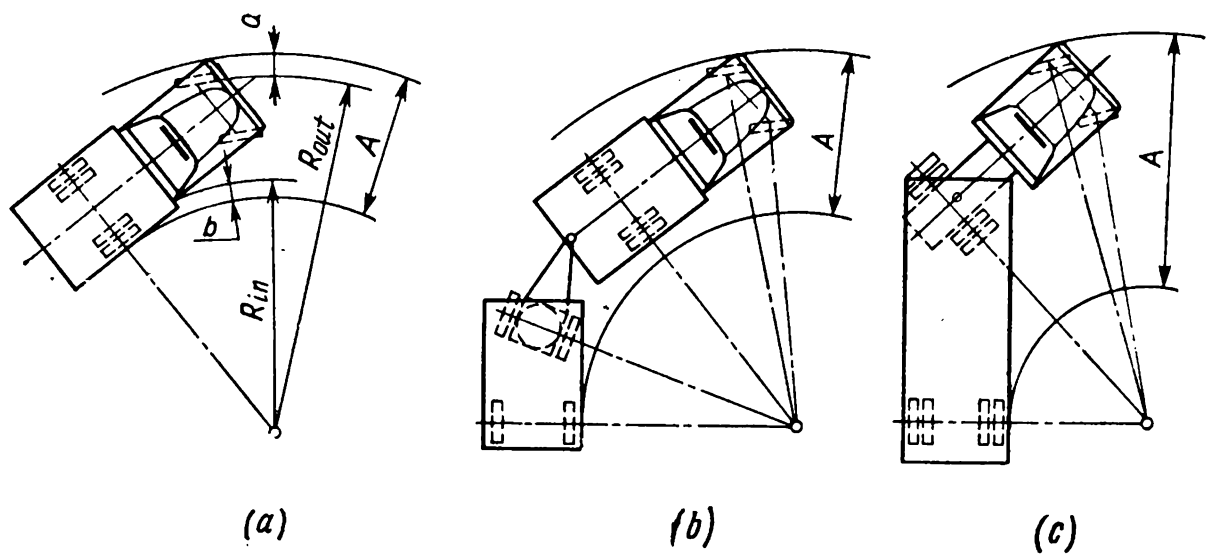


Fig. 15.2. Manoeuvrability parameters
(*a*) single vehicle; (*b*) truck-tractor with a trailer; (*c*) truck-tractor with a semitrailer

front wheel R_{out} (Fig. 15.2*a*), *the width of the traffic path* A occupied by the vehicle during a turn and *the maximum projection of the separate parts of the motor vehicle beyond the path of motion of the outer front and the inner rear wheels* (distances a and b) are manoeuvrability characteristics of a motor vehicle.

The maximum width of the traffic path A (in m) can be found from the formula:

$$A = R_{out} - R_{in} + a + b$$

where R_{in} is the minimum radius of turn of the inner rear wheel in m.

The most manoeuvrable are single vehicles with all the wheels controlled. When trailers are towed, the manoeuvrability of the motor vehicle is slightly impaired. This can be attributed to the fact that, when the trailer train makes a turn, the trailer shifts

towards the cornering centre (Fig. 15.2*b*, *c*) and, as a result, the width of the train traffic path is larger than for a vehicle without a trailer. The width of the trailer train traffic path increases as more towed trailers are attached, and as the trailer base length and the pole length are increased.

15.5. Supporting and Tractive Characteristics of Trafficability

Bad roads and virgin soil offer an increased resistance to the motion of motor vehicles. In order to overcome this resistance, vehicles operating under such conditions must posses better dynamic properties, i.e., a higher tractive force or a larger dynamic factor than vehicles running on good roads. For this reason, the magnitude of the dynamic factor serves as one of the principal characteristics of the vehicle cross-country ability.

In order that the dynamic factor may have the magnitude needed to overcome the maximum resistance to motion, a supplementary gearbox is incorporated in the transmission of a cross-country vehicle. The dynamic factor can also be increased by reducing the weight of the vehicle, which in this case means a smaller load.

The dynamic factor as to the road grip can be found from the formula:

$$D_{gr} = \frac{\varphi_x G_{dr.\,w}}{G_a} \cos \alpha$$

where α = angle of road ascent

$G_{dr.\,w}$ = adhesion weight, i.e., part of the vehicle weight taken by driving wheels, N

G_a = weight of automobile with full load, N

The adhesion weight can be augmented if the number of the driving wheels is increased or the centre of gravity displaced towards the driving axle.

For the cars and trucks with front or rear driving wheels, the ratio $\frac{G_{dr.\,w}}{G_a}$ lies within the limits:

Cars	0.45-0.5
Trucks	0.65-0.75

The maximum value of the dynamic factor D_{max} for these vehicles is within 0.25-0.35. For trucks with all the wheels driving, the ratio $\frac{G_{dr.\,w}}{G_a}$ is equal to unity and the value of D_{max} is within 0.6-0.8.

Specific pressure of the wheels on the road is the main criterion showing the vehicle running ability on roads with soft covering

(noncompact soil, snow, sand, etc.). The specific pressure (in N/m²) can be determined from the formula:

$$P_{sp} = \frac{G_w}{F_w} \tag{15.1}$$

where G_w = weight of the vehicle taken by the wheel, N
F_w = area of contact of the wheel with the road, m²

The area of contact F_w is estimated on a level site with a hard surface from the area of print left by a painted section of the tyre tread on a piece of paper after the suspended wheel has been lowered onto the paper. When determining F_w, the full load should be applied to the vehicle, and the air pressure in all tyres should be nominal. In the case of noncompact soil, the area F_w is estimated by the same method from the horizontal projection of the print of the tyre tread.

The wheel sinks into soft soil until the specific pressure of the wheel on the road becomes equal to the supporting capacity of the soil. The more is the wheel immersed into the soil, the higher resistance is offered to rolling. The force of resistance to rolling may increase so much that the vehicle fails to overcome it by the adhesion conditions. The specific pressure of the wheels on the road can be reduced by decreasing the air pressure in the tyres or increasing the profile and diameter of the tyres and also the number of axles and wheels. In view of this, cross-country vehicles are provided with special tyres of larger diameter and profile in which the air pressure can be changed in motion within broad limits (from 0.05 MN/m² on soft soil to 0.3 MN/m² on roads with a hard surface). The specific pressure on the soil can likewise be reduced by the use of broad-profile and arched tyres or pneumatic rollers.

Figure 15.3*a* illustrates the change in the area of contact F_w of the tyre 12.00-18 depending on the tyre pressure p_t when it rolls on virgin snow. As the tyre pressure is decreased, the area of contact grows larger and the pressure drops to 0.07-0.08 MN/m².

Figure 15.3*b* shows the graphs of change in the force of resistance P_{rd} to motion of a type 6 × 6 vehicle with 12.00-18 tyres with various values of tyre pressure p_t. It is of interest that the forces of resistance to motion on sand and boggy land reach their minimum at tyre pressure of 0.075 MN/m² and on snow at a pressure of 0.15 MN/m².

The pressure of the front and rear wheels of a vehicle on the road is not the same. Since the front wheels roll on less dense soil than the rear ones, i.e., they form the track (compact the soil) before the rear wheels, the specific pressure of the front wheels on the road should be 20-30 per cent less than that of the rear wheels. If their specific pressures are the same the front

wheels will sink into soft soil. In this case, the running ability can be improved if the specific pressure of the front wheels is reduced by decreasing the air pressure in their tyres or by a corresponding redistribution of weight between the vehicle axles.

When a motor vehicle runs on moist soft soil the driving wheels are liable to skid due to sliding or shearing the soil by the tyre cleats. In this case, a lower specific pressure may impair motion because the driving wheel poorly squeezes out the moisture from

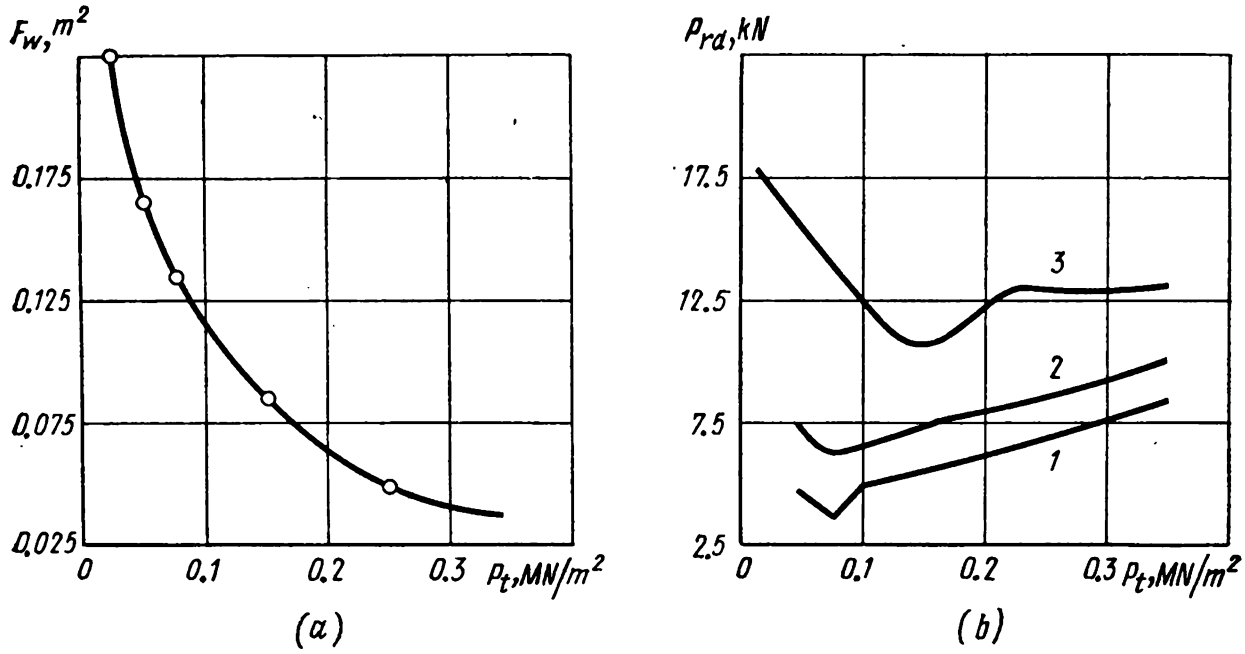


Fig. 15.3. Effect of tyre pressure
(*a*) on the area of contact with the road; (*b*) on the resistance to motion of the vehicle; *1*—sand; *2*—boggy land; *3*—wet virgin snow

the tyre and surface contact areas and begins to skid earlier. Therefore, the specific pressure of the driving wheels should be increased to improve motion on moist soft soil. At the same time, it is better to reduce this pressure in order to eliminate skidding when the soil is shorn off. The maximum tractive force here (in N) is determined from the formula:

$$P_{t\,max} = \sigma F_w \qquad (15.2)$$

where σ is the shearing stress in the soil, N/m^2.

If the shearing stress exceeds the magnitude which can be endured by the soil, the soil is shorn away and the wheel slips forming a deep track.

The contradictory requirements to the magnitude of the specific pressure, when a motor vehicle moves on moist soft soil, can be satisfied in part if large cleat tyres are employed. Due to a small area of contact between the tyre and the supporting surface, before the wheel (cleats) sinks into the soil, the specific pressure of the wheel is large and the moisture is effectively squeezed out

from the area of contact. Immersion of the wheel into the soil increases the total shear area and this reduces the shear stress.

To improve the motion of a motor vehicle on plough-land, strongly soaked road, sand and snow, it is necessary to employ tyres with an especially broad profile and a low air pressure. These are called *arched* tyres because their cross section resembles

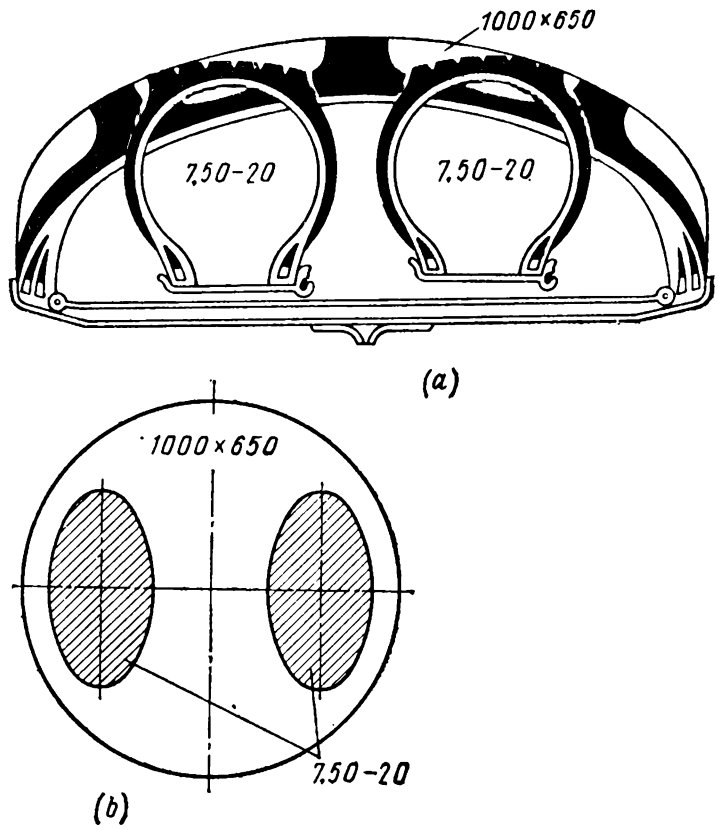


Fig. 15.4. Ordinary and arched tyres
(*a*) comparing the cross section of arched 1000 × 650 tyre and ordinary 7.50-20 tyres of twin wheels; (*b*) area of contact of arched 1000×650 tyre and 7.50-20 tyres of twin wheels

an arch. Fig. 15.4*a* illustrates an arched 1000 × 650 tyre used instead of twin 7.50-20 tyres. The width of the arched tyre profile is 2-2.5 times larger than in ordinary tyres. The low air pressure in arched tyres ensures a considerably larger area of contact with the road as compared with ordinary tyres. It is clear from the comparison of the areas of contact of arched and ordinary tyres on a hard surface that the area of contact of an arched tyre is 2.5-4 times larger than that of ordinary twin tyres replaced by one arched tyre. For example, the area of contact of an arched 1000 × 650 tyre (Fig. 15.4*b*) constitutes 1980 cm^2 and the total area of contact of two 7.50-20 tyres, if used instead, only 500 cm^2.

It has been established by tests that a vehicle with arched tyres runs easily on roads on which a vehicle with ordinary tyres cannot travel. The shortcomings of arched tyres are their smaller

service life as compared with ordinary tyres and a certain drop in load-carrying capacity.

Pneumatic rollers make a vehicle travel better on snow, loose sand and plough-land. With a very wide profile they have a comparatively small outer diameter and a very small inner (fitting) diameter. Pneumatic rollers are designed to give a very high deformation which can reach one-third of the profile height at nominal loads. The low air pressure in a pneumatic roller and broad profile ensure an appreciable contact area and a good road grip which permits high tractive forces to be developed. The pressure exerted by such tyre on the road is within 0.02-0.07 MN/m^2. The considerable volume of air in the pneumatic rollers and their high elasticity sometimes allow the designer to dispense with elastic elements in the suspension.

Broad-profile tyres are subdivided into *universal, high economy and off-the-road tyres.*

The universal tyres are intended for use instead of two ordinary tyres for the twin wheels on the rear driving axle of vehicles of limited cross-country ability and operating on good and bad roads. In this case, the front wheels carry ordinary tyres.

The high economy tyres are intended for vehicles which travel mainly on hard surface roads. They save 20-25 per cent of the tyre material. The tread of wide cross-section tyres of high economy is less thick than in corresponding universal tyres, while the tyre pressure is higher and the deformation smaller.

The off-the-road tyres for motor vehicles with a centralized system of tyre pressure control permit a short-time pressure drop to 0.05 MN/m^2 when travelling over a difficult terrain. The profile of these tyres differs from that of ordinary ones having a tyre pressure control.

For comparison the values of the height H_t of the tyre profile to its width B_t for the different-type tyres are illustrated below:

Ordinary (general-purpose)	0.9-1.1
Wide cross-section of high economy . .	0.75-0.85
Wide cross-section universal	0.6-0.75
Wide cross-section off-the-road with a centralized system of pressure control	0.55-0.80
Arched	0.3-0.4
Pneumatic rollers	0.20-0.35

The pressure p_{sp} (in MN/m^2) of the wheels of modern motor vehicles on an asphalt road is as follows:

Cars, type 4×2	0.17-0.35
Trucks, type 4×2	0.18-0.55
Trucks, types 4×4, 6×4 and 6×6, without centralized system of tyre pressure control	0.2-0.4

The gripping force of the tyres with the road is approximately proportional to the length of their contact with the road. In the case of an ordinary tyre, nearly 8 per cent of its circumference is in contact with the road and up to 16 per cent in a tyre with regulated pressure. However, at a minimum pressure the latter has a limited service life and is appreciably affected by losses due to rolling resistance.

The cross-country ability of a vehicle can be improved by increasing the diameter of the wheels. Thus, cross-country vehicles carry wheels 1.5-2 m in diameter, and 3 m in diameter on certain special vehicles. It should be noted, however, that wheels 17 m in diameter installed on the Canadian cross-country vehicle "Mammoth" did not yield any noticeable advantages. Large wheels inevitably complicate the design and operation (narrow frame for the wheels to turn, special lifting fixtures to replace the tyres).

The tyre road grip coefficient φ_x determines the trafficability of the vehicle moving on moist soil and slippery (icy) road. The vehicle motion on a slippery road can be improved not only by the above methods but also by increasing the grip coefficient with the aid of antiskid chains and blocks mounted on the driving wheels. The grip coefficient is appreciably affected by the pattern of the tyre tread. All-terrain vehicles employ tyres of the type shown in Fig. 7.12*e*, *f* and *g*.

For driving on virgin soil, the vehicle carries tyres with a coarse tread pattern and widely spaced cleats. This tread is less clogged by dirt.

When a vehicle is used on sand it is more efficient to employ tyres with low cleats, and on slippery road, tyres with a markedly opened-out tread pattern (see Fig. 7.12*e*, *g*). If a vehicle has to move on dirt roads the greatest advantage can be derived from tyres with a checker-board tread pattern (see Fig. 7.12*c*, *d*).

The grip coefficient φ_x also depends on the tyre pressure. For example, when the pressure drops from 0.3 to 0.05 MN/m^2 the grip coefficient on soft soil increases from 0.17 to 0.48. This can mainly be attributed to the larger area of the shorn-off soil, i.e., the area enclosed between the cleats of the tread.

15.6. Effect of a Motor Vehicle Design on Its Cross-Country Ability

Effect of the Design of Driven and Driving Wheels

The driven wheels overcome vertical obstacles worse than the driving wheels. This can be explained by the fact that a driven wheel simply runs against an obstacle, while a driving one strives

to climb over it. Fig. 15.5*a* shows the diagram of forces acting on the driven front wheel of a motor vehicle when it climbs over a vertical obstacle with a height *h*.

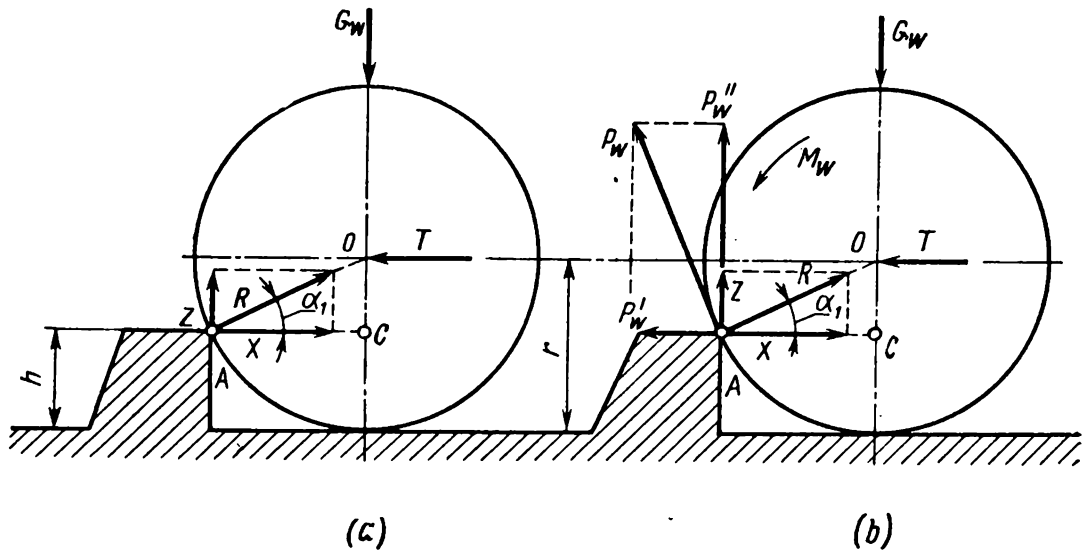


Fig. 15.5. Forces acting on the front wheels when climbing over a vertical obstacle
(*a*) driven wheel; (*b*) driving wheel

The forces acting on the wheel are designated as follows:

T = pushing force taken by the front wheel from the vehicle frame, N

R = reaction of the obstacle, N

Z and X = vertical and horizontal components of reaction of the obstacle, N

From the condition of wheel equilibrium it follows that

$$Z = G_w; \quad X = T$$

The forces acting on the wheel are interconnected by the equalities

$$Z = X \tan \alpha_1 = T \tan \alpha_1; \quad G_w = T \tan \alpha_1 \text{ or } T = G_w / \tan \alpha_1$$

Let us find the value of $\tan \alpha_1$ from the triangle AOC:

$$\tan \alpha_1 = \frac{OC}{AC} = \frac{r - h}{\sqrt{2rh - h^2}} \tag{15.3}$$

hence

$$T = \frac{G_w}{\tan \alpha_1} = G_w \frac{\sqrt{2rh - h^2}}{r - h}$$

It follows from this formula that when $h = r$ the force T grows infinitely large, i.e., when nondriving front wheels run against an

obstacle with a height $h = r$ the motor vehicle cannot overcome it even if the tractive force on the driving wheels is maximum.

Besides the forces T and G_w, the driving front wheel is acted upon by the moment M_w and this generates the force P_w (Fig. 15.5*b*). Let the force P_w be resolved into horizontal P'_w and vertical P''_w components. The action of the forces T and G_w pro-

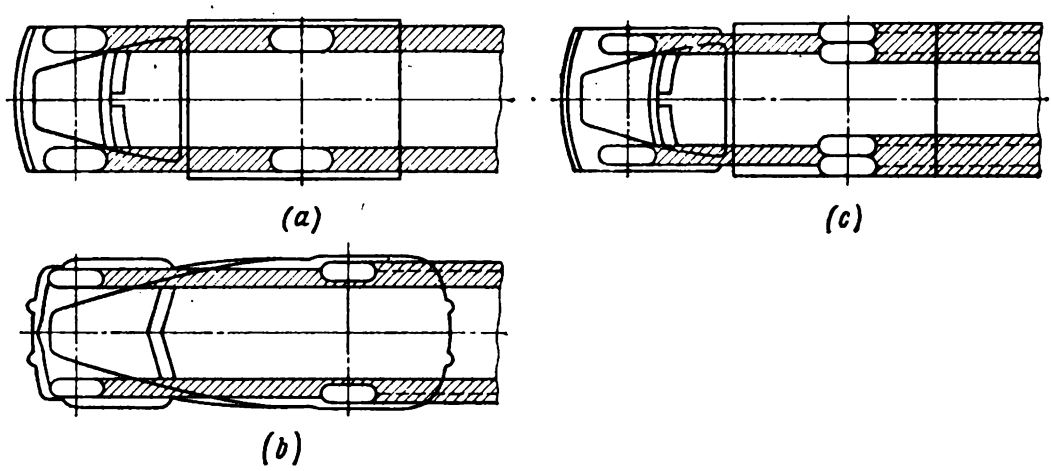


Fig. 15.6. Various mutual positions of the tracks of the front and rear wheels
(*a*) tracks coincide; (*b*) tracks do not coincide; (*c*) twin rear wheels form broader tracks

duces reactions X and Z. Projecting all forces onto the vertical and horizontal axes we get

$$T = X - P'_w; \quad G_w = Z + P''_w$$

The additional force P''_w permits the driving wheel to overcome an obstacle with a height equal to the radius of the wheel, while the force P'_w tends to diminish the component of the force of resistance to motion X.

The track formed as a motor vehicle moves on soft soil produces a considerable resistance to wheel rolling. Therefore, when the tracks of the rear wheels do not coincide, the resistance to motion is higher than if they do coincide. Misalignment of the tracks may be observed in vehicles with single wheels (Fig. 15.6*a* and *b*) and in vehicles with front single and rear twin wheels (Fig. 15.6*c*).

It has been established experimentally that the difference between the size of the front and rear track of single wheels should not exceed 25-32 per cent of the width of the tyre profile. If this difference is larger the tractive properties and cross-country ability of a motor vehicle are reduced by 10-15 per cent.

Effect of the Suspension Design

Motion of motor vehicles, types 6 × 4 and 6 × 6, across country, when the wheels are not detached from the soil, can be limited by the maximum permissible misalignment of their axles, which depends on the type of suspension. In the case of independent and balanced suspensions, the misalignment becomes larger. This improves cross-country ability because the wheels adapt themselves better to the road roughness.

Effect of the Differential Gear Design

The differential gear allows the driving wheels to rotate at various angular velocities.

When the internal friction is small the differential gear distributes the torque equally among the axle shafts, the torque realization being limited by skidding of the driving wheel which lies on the soil with a lesser grip coefficient.

However, the simple differential gear noticeably impairs the cross-country ability because in this case the tractive force on the driving wheels is determined by the wheel which grips the road the less. For this reason the tractive force may prove not enough to overcome the resistance to motion.

The differential gear distributes the torque (in N·m) between the driving wheels as follows:

$$M_1 = 0.5\,(M + M_{fr\,dif}); \quad M_2 = 0.5\,(M - M_{fr\,dif})$$

where M_1 and M_2 = torques on the driving wheels which rotate slower and faster, respectively, N·m

$M_{fr\,dif}$ = moment of friction in the differential gear produced by the relative motion of its parts, N·m

M = torque on the driven pinion of the main gear, N·m

From the viewpoint of vehicle cross-country ability friction in the differential gear is useful as a larger torque can be transmitted to the nonskidding wheel and a smaller one, to the skidding wheel, thus preventing the skidding. The total tractive force on two driving wheels (in N) reaches its maximum:

$$P_{t\,max} = 2P_{gr\,min} + \frac{M_{fr\,dif}}{r}$$

where $P_{gr\,min}$ is the tractive force on the wheel with a smaller grip in N.

Friction in a simple differential gear is not high in view of which the total tractive force increases only by 4-6 per cent. Fric-

tion is much larger in the worm and cam differentials installed in cross-country vehicles and the total tractive force is therefore increased by 10-15 per cent. *Increased friction differentials* are also frequently used. The design of one of such differentials is

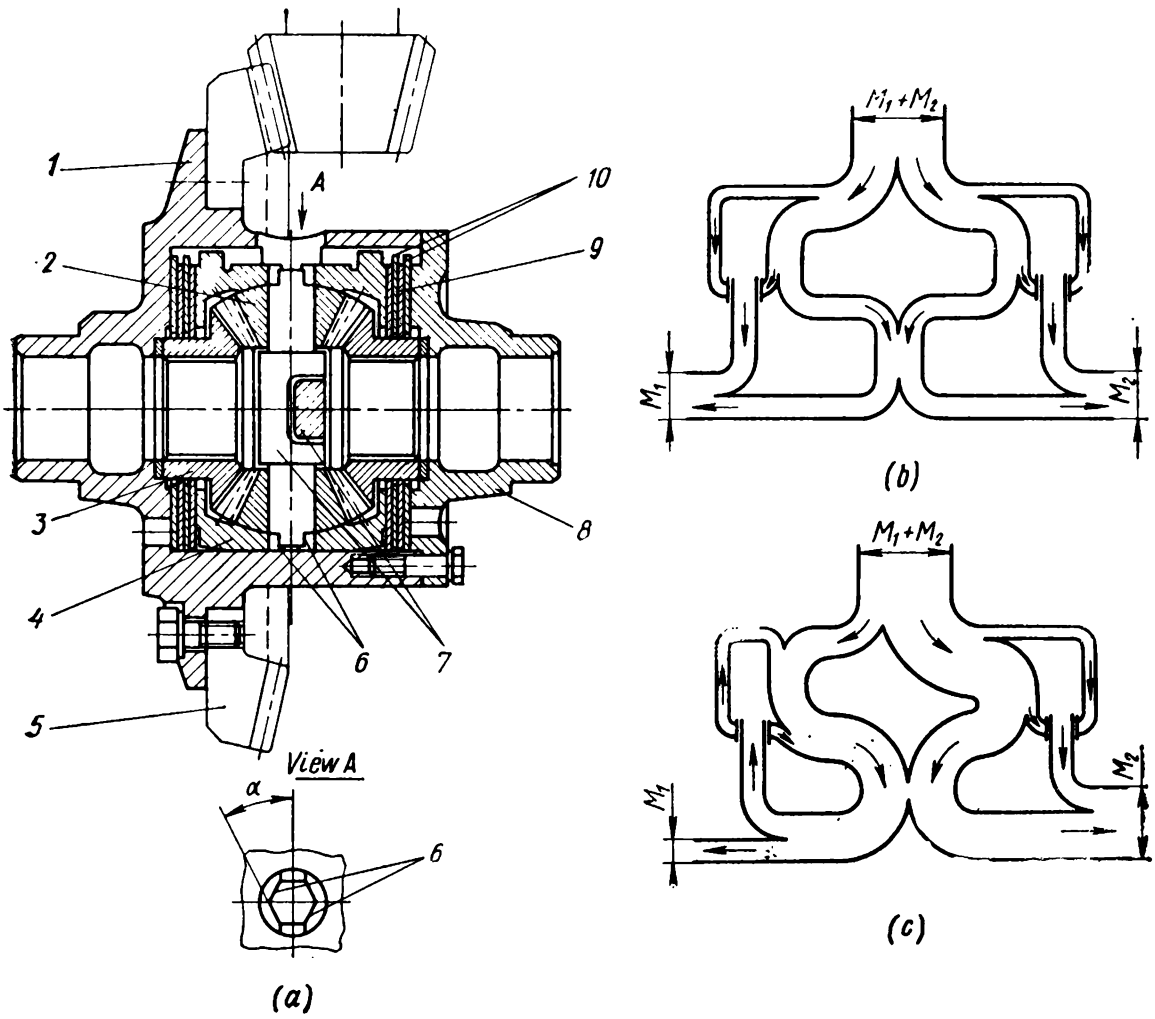


Fig. 15.7. Design of self-locking differential gear and distribution of torques according to a simplified diagram

(*a*) design of differential gear; (*b*) straight motion (torques on axle shafts $M_1=M_2$); (*c*) cornering or motion at various wheel grip coefficients (torque $M_1 < M_2$); *1*—differential casing; *2*—planet pinion; *3*—semiaxle pinion; *4*—pressure disk; *5*—driven pinion of the main gear; *6*—chamfers on differential axle and pressure disk; *7*—differential axles; *8*—cover; *9*—disks with inner splines; *10*—disks with outer splines

shown in Fig. 15.7. It is provided with two friction clutches whose disks *9* are connected to the half-shaft gears of the differential, and disks *10*, to housing *1*.

The packet of the clutch disks is compressed automatically because the axles *7* of the differential and pressure disks *4* are provided with chamfers, the arising axial effort being proportional to the torque transmitted by the differential gear.

The friction clutches prevent the change in the angular velocity of one of the driving wheels, and the moment of friction of the differential gear depends on the received torque, the number of

disks, the chamfer angle α on the axles of the differential and the pressure disks. During straight motion (Fig. 15.7*b*), the torque is transmitted to the axle shafts in two ways: through the bevel gears of the differential and through both friction clutches, the axle shaft torques being mutually equal ($M_1 = M_2$). When cornering (see Fig. 15.7*c*) the torques are redistributed and a larger torque ($M_2 > M_1$) is supplied to the right-hand driving wheel.

The use of differentials with forced locking (differential disengaged) allows the tractive force to be increased by 200-300 per cent. However, when the differential is locked, the tractive properties of a vehicle are improved only with a large difference between the grip coefficients on sections of the road beneath the left-hand and right-hand driving wheels or when the load is reduced on one of them during motion on a very uneven surface. But such cases are observed rather rarely. When the differential gear is locked during motion in real road conditions the total tractive force increases by 20-25 per cent because the difference between the grip coefficients on the portions of the road under the right- and left-hand driving wheels is comparatively small.

The self-locking differential gears with free-wheeling clutches are distinguished by the torque distribution which ensures the maximum possible tractive force (the differential is disengaged) at any ratio of the driving wheel grip coefficients. A self-locking differential improves the vehicle motion on slippery soil because it acts automatically and easily overcomes this section when one of the driving wheels rolls on the road with a low grip coefficient.

The maximum tractive force on the driving wheels with a self-locking differential

$$P_{t\,\max} = \varphi_{\max} \frac{P_t}{2}$$

where $\varphi_{\max}$ is the maximum grip coefficient for one of the driving wheels.

Effect of Tyre Pressure Control

At present, use is made of a single- or double-conduit centralized system of tyre pressure control.

By way of example, Fig. 15.8 illustrates the *single-conduit system* employed in ЗИЛ motor vehicles. This system consists of compressor *1*, air cylinder *2* with a safety valve *3* adjusted for a maximum excess pressure of 0.6 MN/m^2, central cock *6* and assembly *4* of tyre cocks to control the system, pipes *9* and *11*, flexible hoses *10* and air let heads *5*.

Central cock *6* has a delivery and an exhaust valves.

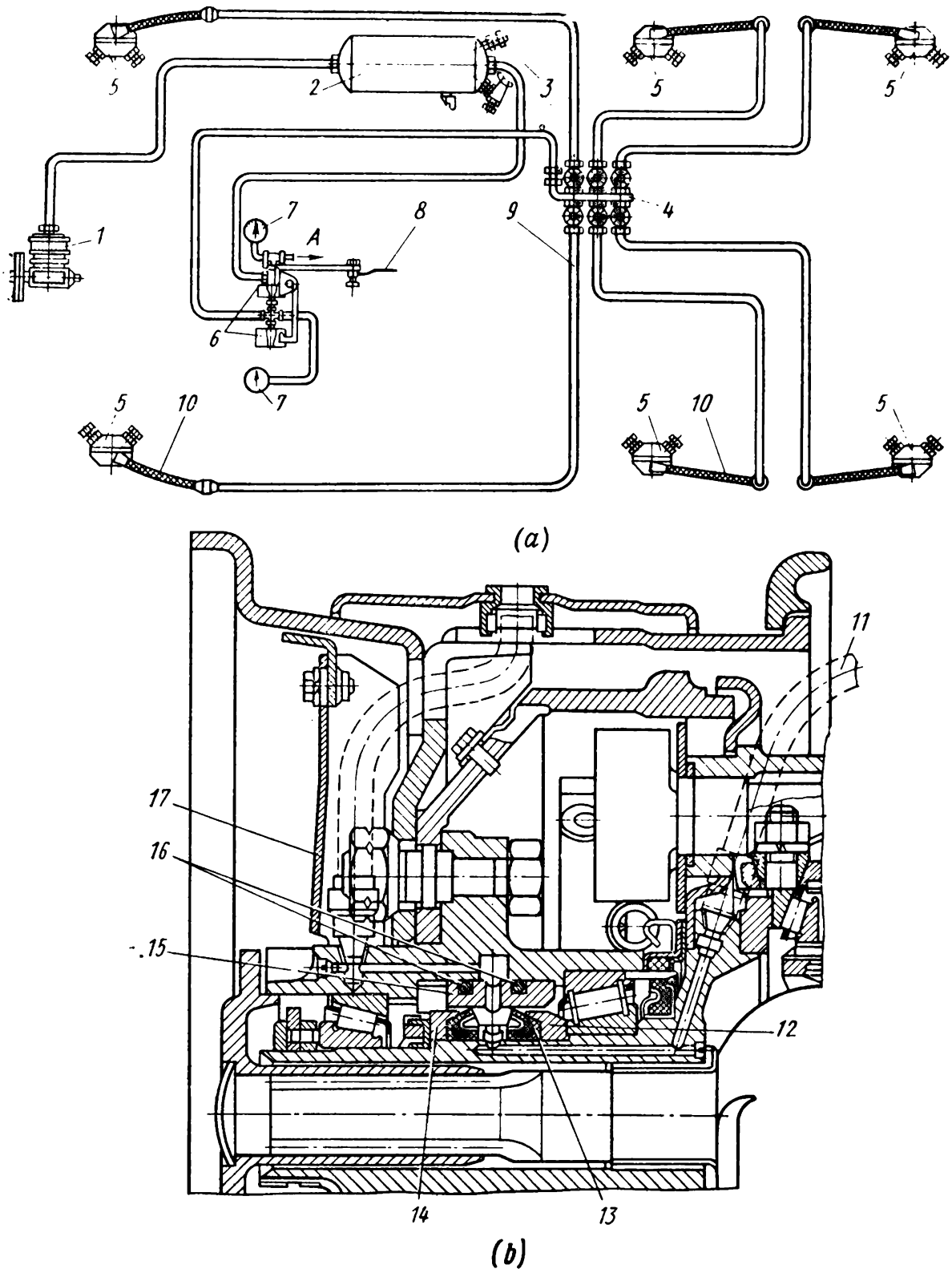


Fig. 15.8. Single-conduit system of tyre pressure control:

(*a*) diagram; (*b*) air inlet head; *A*—to brakes; *1*—compressor; *2*—air cylinder; *3*—safety valve; *4*—tyre cock assembly; *5*—air inlet head; *6*—central cock; *7*—manometer; *8*—valve control lever; *9* and *11*—pipes; *10*—flexible hose; *12* and *14*—holders; *13*—collar; *15*—bushing; *16*—sealing rings; *17*—protective shield

The valves are controlled by lever *8* on the instrument panel. When the lever is in the neutral position the valves are closed, the air cylinder is disconnected from the tyre cock assembly and the assembly itself, from the atmosphere. When the central cock is shifted to the pumping position it cuts off the tyre cock assembly from the atmosphere and links it with the air cylinder. With the cock in the release position the tyre cock assembly is disconnected from the air cylinder and communicates with the atmosphere.

The tyre cock assembly has valves with handwheels equal to the number of the vehicle wheels. The valves cut off air supply to the tyres. If all the valves are opened air can be delivered into all tyres and exhausted at the same time. In this case the air pressure will be the same in all tyres because they are interconnected. By opening the valves in succession and setting the central cock in one of the above positions it is possible to expel air from individual tyres, pump it into the tyres or check the pressure by means of manometer *7*.

The air inlet head is shown in Fig. 15.8*b*. Air is fed along pipes *11* of the pressure control system via an inclined and an axial hole in the steering knuckles and in the housings of the middle and rear axles. The annular gap between stationary and revolving parts is sealed by collars *13* mounted in holders *12* and *14*. A nut is used to adjust holder *14* in the axial direction and press the collars against a spacing sleeve and the inner race of the wheel bearing to make the collars fit tightly against bushing *15*. Compressed air is delivered along pipe *11* into the inclined and axial holes in the steering knuckle, then through the annular chamber between the collars, the radial holes in bushing *15*, the annular groove and axial hole in the wheel hub it is admitted into the wheel tyre through a flexible hose protected by shield *17*.

15.7. Self-Recovery Devices

Some cross-country vehicles carry winches, operated from a power take-off gearbox, which are used to overcome very difficult sections of the terrain and ascents and also for self-recovery of the stuck-in motor vehicle. Winches with a horizontal pulling drum are most popular and with a vertical drum, less popular. As a rule, the winches are installed in the front of small and medium load-carrying capacity motor vehicles. A winch thus located has a simple drive and is easy to service but the overload on the front axle, reduced front angle of trafficability and difficult starting with a crankhandle are among its shortcomings.

The winch drum drive usually includes a worm reduction gear. A brake is employed to hold the drum under load. In order to protect the elements of the winch from damage in the case of

overload, the drive taking the torque from the gearbox is provided with a torque limiter. The simplest limiter is a shear pin which breaks when the torque exceeds the permissible value. The length of the winch rope should be about 80-110 m. The mean operating

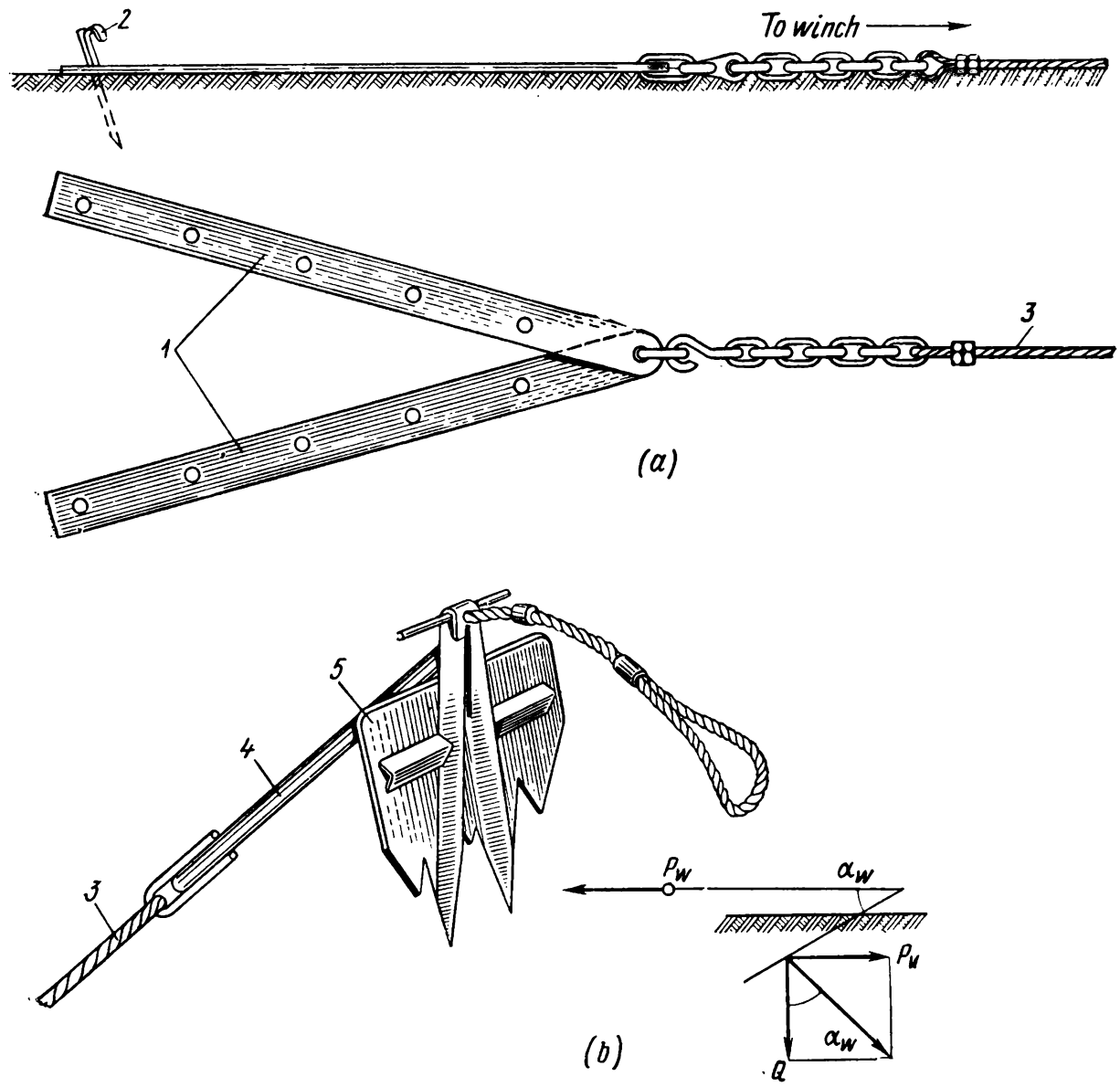


Fig. 15.9. Anchoring devices
(*a*) flat folding anchor; (*b*) self-deepening anchor; *1*—bar with holes; *2*—pintle; *3*—winch rope; *4*—pull rod; *5*—fluke

rope winding speed in low gears and with a low engine crankshaft speed is within 0.15-0.5 m/s. In heavy trucks, the winch is frequently accommodated at the vehicle centre.

Before self-recovery of the motor vehicle carrying a winch the end of the rope is attached to a support which can endure the maximum pulling force developed by the winch. Stumps, trees and other local objects are employed as natural support points. If they are not available use is made of various *anchoring devices.* The simplest is a corkscrew-type anchor screwed into the soil by

a crow-bar. The anchor is a steel bar 10-16 mm in diameter and up to 1 m long. It can be employed to pull out low-capacity vehicles. The rope of the winch is secured in the eye of the bar.

A flat folding anchor (Fig. 15.9*a*) can be utilized on compact soil. It occupies little space and can conveniently be carried in the body of the truck. The lengths of bar *1* and pintle *2* depend on the load carrying capacity of the vehicle. Thus, with a capacity of 2.5-3 tons the bar is 1.2 m long and the pintle, 0.2 m long.

In these anchoring devices the surface of contact between the cylindrical pintles and the soil is not large. In order to make an anchoring device sustain a considerable pulling force of the winch, many pintles have to be driven into the soil. The structure becomes cumbersome and more time is required to install it. A much greater bearing surface is offered by a *self-deepening anchor with a fluke 5* (Fig. 15.9*b*). Under the action of the pulling force taken by rope *3* and pull rod *4* the fluke of such an anchor tends to penetrate deeper into the soil. If a force P_w is applied to the fluke the force Q which strives to immerse it into the soil is equal to the product $P_w \cot \alpha_w$. In this case the pressure on the soil will be (in N/m²).

$$q_{soil} = \frac{P_w}{F_{fl} \sin \alpha_w}$$

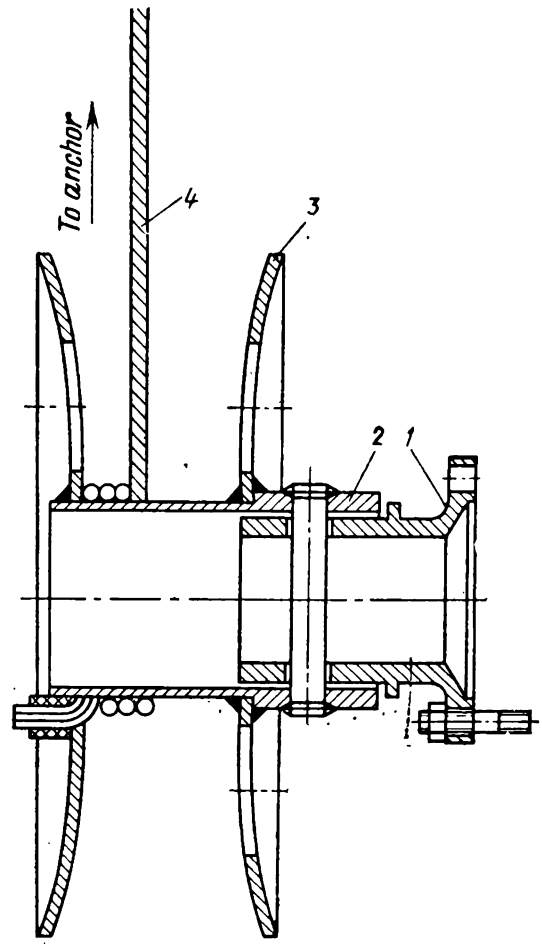


Fig. 15.10. Winch mounted on the driving wheel
1—flange; *2*—drum hub; *3*—drum; *4*—rope

where F_{fl} is the area of the bearing surface of the fluke plunged into the soil, m²

If a vehicle has no winch self-recovery can be done by the driving wheels carrying *self-recovery winches* on their hubs (Fig. 15.10). Flanges *1* of the self-recovering device are permanently secured to the axle shafts by the pins and nuts fastening the driving wheels. To mount the winches on the wheels, drums *3* with ropes *4* are placed with their hubs *2* onto flanges *1*. The ropes are unwound and their ends are attached to the supports nearby. The total pulling force on the ropes of the self-recovering arrangement exceeds the pulling force on the driving wheels of the vehicle by the number of times by which the wheel radius *r*

is larger than the drum radius r_{dr} and attains 70-80 per cent of the vehicle weight, the force enough to pull out a stuck vehicle even on difficult sections.

Fixtures Improving the Cross-Country Ability

When motor vehicles operate in arduous road conditions wide use is made of chains of various design which increase the gripping surface of the wheels, i.e., ensure a higher tractive force as to the grip conditions. The chains are fastened to the vehicle driving wheels.

The simplest fixture increasing the grip of wheels is a *twisted chain* with end locks. When the distance between the wheel disks and the brake drums is wide such chains can be wound uniformly over the rims and tyres and then secured by the locks in twenty minutes. *Bracelet-type chains* are placed on the rim and tyre of one or two wheels (with twin wheels). Such chains, on compact snow and icy roads and also on soaked dirt roads with a hard base, can increase the tractive force by 20-45 per cent as compared with tyres without a developed tread pattern. Such chains should not be installed in the case of weak soil (loose sandy and boggy soil, virgin snow) because the shear of soil by the projections of the chains may cause the wheels to plunge rapidly into the soil and make the vehicle stick.

When a motor vehicle moves on deep loose snow and in slush its motion can be improved by employing antiskid *track chains.* Fig. 15.11 shows a bracelet-type track chain for a motor vehicle with one driving axle. The chain is composed of track shoes *1* made of *T*-shaped steel or steel angles welded together, linking *2* and fastening *3* chains and fastening bolts *4* with nuts. The track shoes act as cleats and have guiding flanges *5* which prevent the chains from dropping during cornering. An angular arrangement of the chain track shoes as to the wheel axles improves the smoothness of run of the vehicle and self-cleansing of the chains, and also ensures shockless wheel rolling. When the snow density is 0.3 g/cm^3 a motor vehicle, without chains, can travel on snow up to 10 cm deep. With track chains installed the snow depth which may be traversed increases to 40-50 cm, i.e., by 4-5 times. The shortcoming of such chains is their large mass.

A caterpillar chain (Fig. 15.11) is employed for three-axle motor vehicles. The chain consists of track shoes *1*, with guiding flanges *5*, connected by chains *2* and rings *6*. The two extreme track shoes are linked by tensioning chains *7* and pins *8*. The chain is then tensioned with the aid of a tensioner composed of screw *9* with a left- and right-hand thread and nuts with grips *10*. A three-axle vehicle, without chains, can travel over a snow covering up

to 35 cm deep, and with track chains, up to 70 cm. In this case the improved travel ability can be explained by a drop in specific pressure on the supporting surface and, hence, by the reduction in the depth of the track and the resistance to motion as well as by a better grip of wheels.

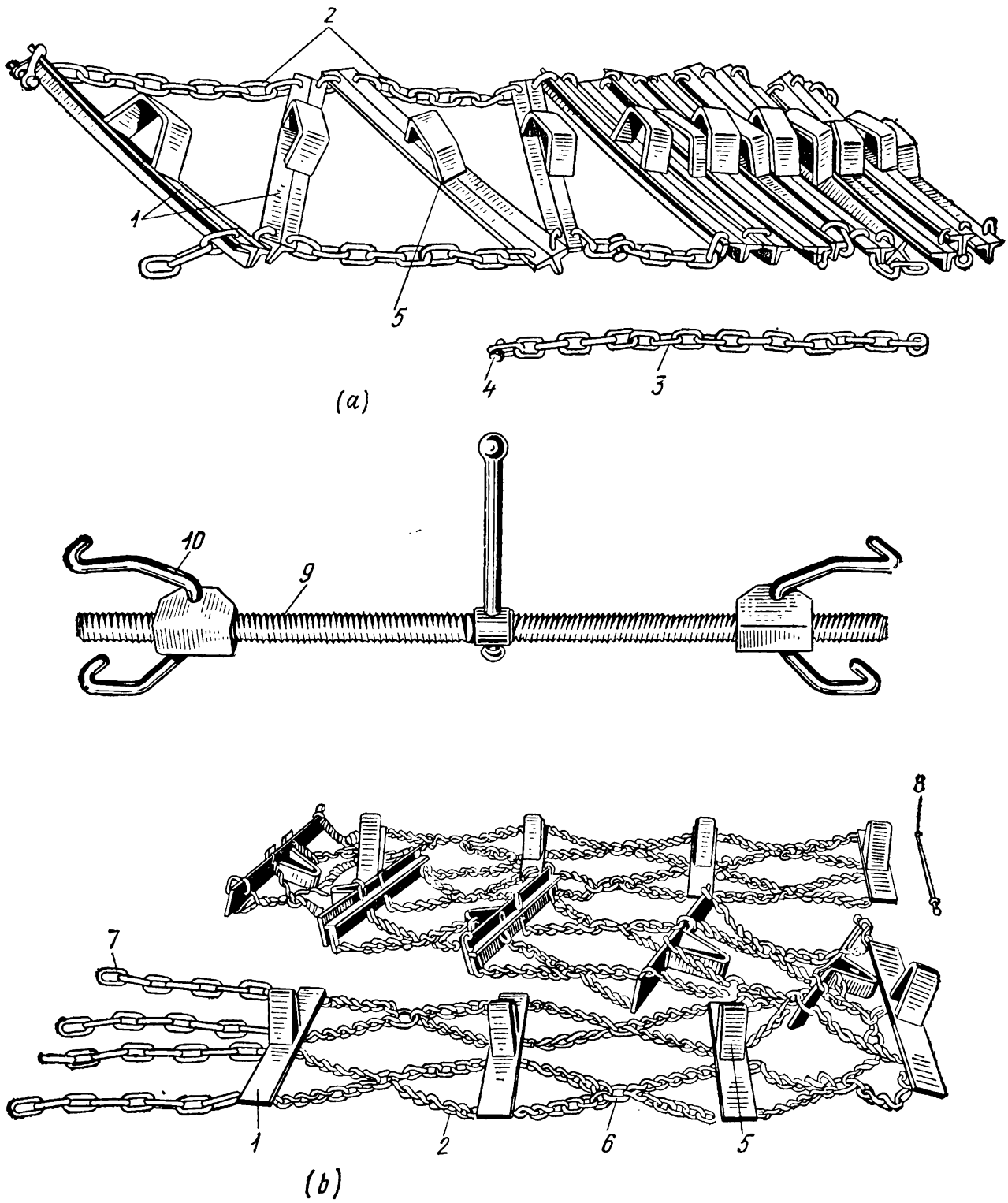


Fig. 15.11. Track antiskidding chains

(*a*) bracelet-type chain; (*b*) caterpillar chain; *1*—track shoes; *2*—linking chains; *3*—fastening chain; *4*—bolts with nuts; *5*—guiding flange; *6*—ring; *7*—tensioning chain; *8*—pin; *9*—tensioner screw; *10*—grip

It should be borne in mind that these fixtures can be utilized only for a temporary increase in motion over heavy terrain, and they should be taken off when traversing roads with a hard surface.

It may occur in driving practice that a motor vehicle gets stuck due to the driving wheel skidding. For a self-recovery of such a vehicle, use is made of *antiskidders*. This simple device may, for example, be a mat woven from ropes 5-6 mm in diameter, 6-7

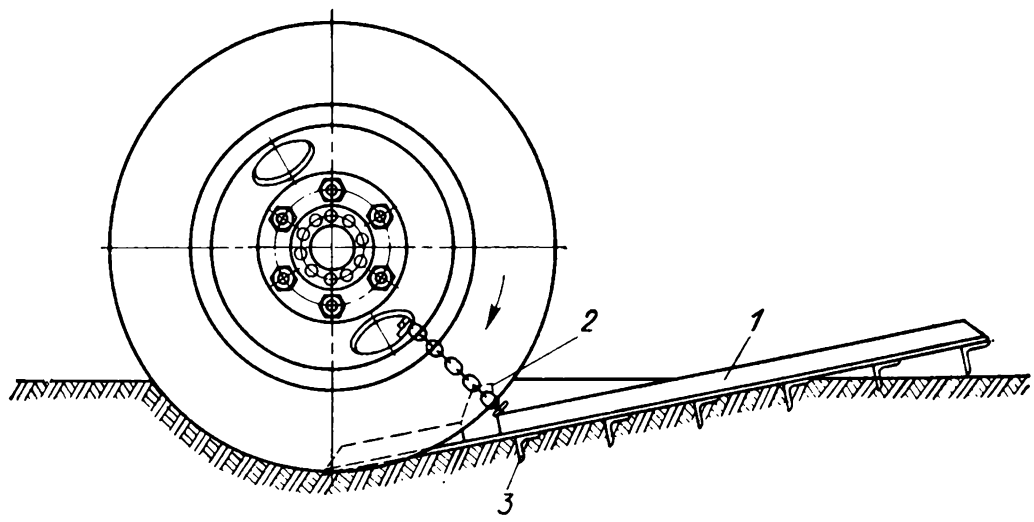


Fig. 15.12. Antiskidder
1—longitudinal angles; *2*—grip; *3*—cross angles

pieces of tied blocks or the fixture shown in Fig. 15.12. The fixture is welded from two longitudinal angles *1*, two grips *2*, cross angles *3* and a chain.

15.8. Special Types of Cross-Country Vehicles

The motor vehicles cross-country ability can be improved by increasing the number of wheels to six, eight and even ten, most of which, and sometimes all, are driving. A multiple-wheel all-drive design complicates the transmission and impairs the vehicle characteristics as a transporting facility. Apart from the main gearbox, such vehicles carry several distributor boxes and inter-axial differentials, and the units of the transmission are linked by several universal-joint drives. This increases the weight so that the latter exceeds the useful load-carrying capacity (by 2-3 times for some vehicles), and the power losses in the transmission during motion on bad roads sometimes reach half the engine power.

Most of the cross-country vehicles have an ordinary-type frame. However, the tendency to use wheels of a large diameter with

the same width of the frame, reduce the weight, improve turnability and trafficability of the vehicle has led to designs with a composite frame. These vehicles are known as *articulated vehicles*. Depending on the principle of turn of the vehicle, they can be designed with a frame articulated with respect to the crosswise horizontal axis, cross and longitudinal horizontal axes, longitudinal and vertical axes or with respect to all the three axes.

Motor vehicles with caterpillar tracks have been produced for over fifty years but their design has not undergone any appreciable changes. They differ from wheeled vehicles in the type of drive. The caterpillars (endless chains) act as rails laid before the vehicle and picked up after. The wheels of caterpillar drive (support rolls) press on the soil through track shoes. Since the area of contact of a caterpillar vehicle is 3-5 times and the length of the supporting surface, 2.2-2.4 times larger than in wheeled vehicles, the cross-country ability of the former is much better. All other conditions being equal, wheels slip more often than caterpillars because slipping depends on the length of the contact area between the tyre and the soil. As compared with a wheel drive, the principal shortcoming of a caterpillar drive is the high internal losses in the drive. At present, light caterpillar drives (band-type, pneumatic, with spaced track shoes) have been designed which ensure good operating ability, better reliability and a longer service life in difficult road conditions.

A tracked vehicle is turned under the action of the turning moment produced by different tractive forces on the left-hand and right-hand caterpillars.

Motor vehicles of limited cross-country ability (general-purpose) can traverse hard-bottomed shallow fords (0.5-0.6 m). The ford depth passable by a vehicle depends on the height of separate parts of the engine which must never enter the water (air strainer, filter of the crankcase ventilation system, spark plugs and ignition distributor).

Broad and deep water obstacles are overcome by *special amphibian vehicles* designed, as a rule, on the base of wheeled motor vehicles, type 4×4 or 6×6.

The body of an amphibian vehicle is made as a water-tight flat-bottomed boat. The vehicle carries a propeller, rudder and water pumping system.

The swimming condition will be observed if the submerged portion of the amphibian vehicle displaces the volume of water whose weight is equal to that of the vehicle. Since water may get into the body of the vehicle traversing a water obstacle, a certain buoyancy reserve must be given. Full buoyancy reserve corresponds to the part of the volume located above the waterline of a water-impermeable body.

Two conditions should be observed for a vehicle in the water to be in equilibrium. Firstly, the force of gravity of the vehicle (directed downwards) and the buoyancy force (directed upwards) should be mutually equal. If the buoyancy force is less or larger than the force of gravity the vehicle will immerse or float higher out of the water until these forces are equalized. Secondly, the force of gravity and the force of buoyancy must act along one vertical line, i.e., the centre of gravity and the *point of application of the buoyancy force* should lie in the same plane. If this condition is not observed the equilibrium will be disturbed and a moment appears which tends to change the vehicle position.

An amphibian vehicle should be *water stable.* This means that if external forces force the vehicle out of the equilibrium position it should right itself and regain its original position after these forces cease to act.

When a vehicle is tilted the position of its centre of gravity does not change, but the point of application of the buoyancy force is displaced. If this point is above the centre of gravity the tilt causes the righting moment, since this point is displaced relative to the centre of gravity towards the tilt. As a result, the moment tends to turn the vehicle in the direction opposite to the tilt until the point of application of the buoyancy force is on one vertical line with the centre of gravity. If this point lies below the centre of gravity the tilt of the vehicle generates an opposite moment because this point is displaced in the direction opposite to the tilt which begins to increase. Thus, *the point of application of the buoyancy force should be above the centre of gravity for an amphibian vehicle to be stable.*

An amphibian vehicle can be tilted with respect to its longitudinal axis (roll) and with respect to the crosswise axis (trim). Distinction is therefore made between *longitudinal* and *cross stability.* In practice greatest importance is attached to the cross stability.

To preserve the stability with respect to the cross axis, the rear of the vehicle should be submerged in the water somewhat deeper than the front portion. Such a trim by the stern constituting 5-8° does not allow the front of the vehicle to submerge by an oncoming wave. The trim by the bow is not permitted because it noticeably decreases the speed of the vehicle and makes its front portion plunge and heavily rise on the wave. The maximum angle of permissible roll of modern amphibian vehicles is within 17-20°.

When a vehicle moves on the water its speed considerably drops as compared to its speed on a hard surface because water offers appreciable resistance to motion. The speed of up-to-date amphibian vehicles in water is 2.5-4 m/s.

The most difficult moments in the operation of amphibian

vehicles are plunging into the water and climbing onto dry land. When entering the water the adhesion weight diminishes because the vehicle tends to come to the surface. As a result the driving wheels begin to slip on soft soil. The process of plunging is much affected by the mass of the amphibian vehicles, its distribution between the axles and the height of the centre of gravity. When selecting the entering place preference should be given to the steeper but not too precipitous banks. Amphibian vehicles of modern design can overcome water obstacles with a maximum bank steepness of 30-35°.

A larger tractive force is required for a vehicle to leave the water than to come in. For this reason, before the vehicle starts to leave the water the driver should engage the drive of the front wheels so that their pulling force is utilized together with the pushing force of the propeller.

For an amphibian vehicle to climb successfully from the water, the angular velocity of the wheels v_w should correspond to the speed of motion v. If the speed v_w exceeds the speed v the driving wheels begin to slip sharply reducing the grip coefficient and, as a result, the vehicle can stick.

When overcoming water obstacles the vehicle motion can considerably be increased if the tyre pressure is regulated according to the riverside soil density and then reduced near the bank to 0.05-0.08 MN/m^2.

An amphibian vehicle is steered by turning the front wheels and the rudder mounted in the rear of the body.

Chapter XVI SMOOTHNESS OF RUN OF A MOTOR VEHICLE

16.1. General

Prolonged body oscillations caused by the vehicle motion excessively fatigue the passengers and driver who begin to feel giddy and experience car sickness detrimental to their health. Also, the driving is less efficient. These oscillations affect the load being transported and the vehicle itself. Smooth running and better comfort during transit are, therefore, among the principal requirements imposed on a modern motor vehicle.

Oscillations are mainly caused by road irregularities of different size and contour even on roads with cement- and asphalt-concrete surfaces. The roughness may be of two kinds: microirregularities (3-5 mm high and 8-10 mm long) and waviness (10-12 mm high and 5-8 mm long).

On roads with intense traffic, waviness in the road usually appears one or two years after the road has been built, which worsens the smoothness of run.

16.2. Testing for Smooth Running

Experimental research into smoothness of run of a vehicle is conducted by means of laboratory and road tests using various methods. Apart from a general estimation of the smoothness of run the tests are also intended to study the effect of such factors as the amplitude, speed, and acceleration of oscillations, oscillating frequency of sprung and unsprung masses, etc.

During the smooth running laboratory tests the oscillations are excited by the following methods:

pulling down the body and quickly letting it or its axles off disengaging the clutches at the same time;

raising the vehicle by means of special platforms to a height of 50-60 mm and rapidly dropping it. (The first two methods are employed to study the oscillating frequencies and the nature of their damping and also to determine the centres of oscillation of the vehicle);

mounting the vehicle wheels on a test stand with rotating drums which have projections or eccentricity (the drums may be arranged under the front or rear wheels or under all wheels of the vehicle; in the former instance oscillations of the body are

generated on one of the suspensions and in the latter, oscillations in coupled circuits appear on the front and rear suspensions);

oscillating periodically the platforms carrying the vehicle;

mounting the vehicle wheels on moving endless bands with irregularities.

When stand tests are conducted use is made of irregularities, 50 mm high and 250, 500 and 1000 mm long, which resemble in profile the typical road irregularities. Besides, short irregularities are also employed — 120 mm long and 25, 35 and 50 mm high, which correspond, for example, to cobblestone and other road surfaces. When oscillations of a motor vehicle are investigated in a laboratory the oscillations of the body, wheels and passengers on their seats are registered by means of a recorder or a light-beam oscillograph. Simultaneously, timing marks are placed on the tape of a recorder or an oscillograph. A tachometer is used to find the angular velocity of the drums from which the speed of the vehicle is calculated.

The data obtained from laboratory tests are processed in order to estimate the oscillating frequency of the sprung and unsprung masses and also the amplitude, speed, and acceleration of oscillations as well as other parameters.

In order to obtain comparable results for different vehicles under road test conditions, oscillations are excited by making the vehicle run over artifical irregularities of definite profile placed on an even road (two irregularities, each for left- and right-hand wheels). The irregularities are traversed at various speeds beginning with 1.4-2.8 m/s, each subsequent speed being in excess of the previous one by 1.4 m/s. During road tests use is usually made of sinusoidal longitudinal profile irregularities. The road is paved with long (2.5-3.5 m) and short (0.8-1.2) irregularities. The time of run at a speed of 5.5-11 m/s over long irregularities is equal to the period of low-frequency oscillations and over short ones, to the period of high-frequency oscillations.

Various vehicles of the same class should be tested on irregularities having identical length and height. In the case of trucks, the irregularities should be 0.8 and 3 m long. When testing vehicles with rigid or soft suspension the height of irregularities should be 40 or 80 mm, respectively. During the tests the absolute motions of the body and unsprung masses are recorded, i.e., the motions of masses oscillating with respect to the road.

After processing the test information the maximum deviations of the parts of the body from a datum line during the first period of oscillations and the frequency of oscillations are determined. The deformation of the elastic element of the suspension can be estimated from the distance between the oscillatory curves of the

axle and the body. Besides, the frequency of oscillations of the unsprung masses is also determined.

The angular oscillations of the body are found from the vertical motion of its points located above the front and rear axles.

During the smooth running tests the speed of the vehicle must be constant. If the speed is measured by a speedometer, errors may reach 10-30 per cent. The accuracy of measuring small vehicle speeds can be enhanced if they are determined with the aid of a tachometer. However, the most accurate estimate of the average vehicle speed is provided by timing marks spaced on a photofilm

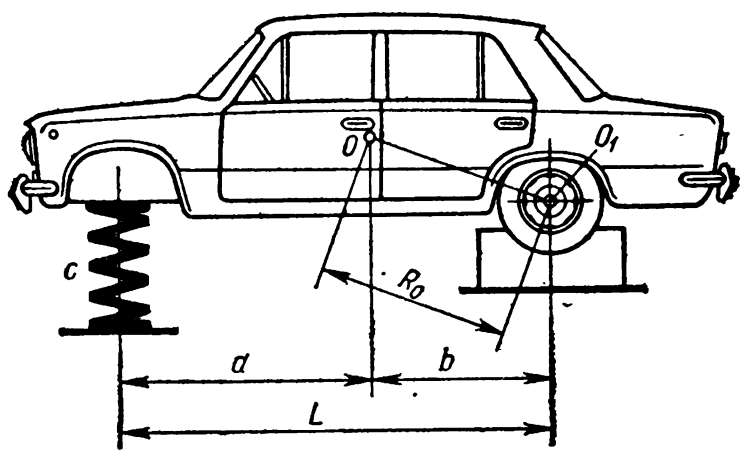


Fig. 16.1. Determining the moment of inertia of sprung masses

or recorder tape with an autotimer. If the length of the road section and the time of motion are known one can find the average speed of the vehicle.

The acceleration values during oscillations are determined from the records put on the tape by an accelerograph.

When investigating the smoothness of run and calculating the suspension it is necessary to know the moment of inertia I_b of the sprung masses (body) with respect to the cross axis passing through the centre of gravity and perpendicular to the longitudinal axis of the vehicle. The simplest method of determining I_b consists in oscillating the vehicle about one of its axles. For this purpose the front or rear wheels, with the elastic elements of the suspension being wedged and the pressure in the tyres increased to the maximum permissible value, are installed in V-blocks as shown in Fig. 16.1. The other axle is removed together with the suspension and replaced by springs. The front of the vehicle is swung to excite oscillations whose frequency is determined by means of a stopwatch or from the records of oscillations by a special instrument — vibrograph. The moment of inertia of the sprung masses of the vehicle I with respect to the axis of swing O_1 can

be found from the formula (in $N \cdot m \cdot s^2$)

$$I_1 = \frac{cl^2T_1^2}{4\pi^2} \tag{16.1}$$

where c = stiffness of the spring, N/m
l = distance from the spring centre to the swinging axis of the body, m
T_1 = oscillatory period, s

The moment of inertia I_b of the sprung masses of a vehicle (in $N \cdot m \cdot s^2$) relative to the cross axis passing through the centre of gravity O and perpendicular to the longitudinal axis of the vehicle can be found from the formula

$$I_b = I_1 - m_b R_0^2$$

where m_b = mass of the sprung parts (body) of the vehicle, kg
R_0 = distance from centre of gravity O of the sprung masses to the swing axis O_1 (see Fig. 16.1), m

The effect of shock absorbers on the smoothness of run is studied on special test stands on which the forces of resistance at various frequencies and oscillatory amplitudes and also at various pressures and temperatures of the fluid are determined.

16.3. Characteristics of the Smoothness of Run

The effect produced by the oscillations of the motor vehicle on the human organism is estimated subjectively. Even today not one single parameter exists which estimates the smoothness of run or helps to establish the quantitative relation between the physiological effect of oscillations on the human organism, the nature of these oscillations and the design features of a motor vehicle. For this reason the smoothness of run is estimated by means of the characteristics described below.

The oscillatory period T which is the time in seconds during which the body performs a full oscillating motion. *The frequency of oscillations* $n = 1/T$ in Hz, i.e., the number of oscillations (cycles) per second.

The oscillatory amplitude z_{max}, i.e., maximum deviation (shift) of the body from the position of equilibrium in m.

The velocity of oscillations equal to the first derivative of motion in time in m/s.

The acceleration of oscillations equal to the second derivative of motion in time or the first derivative of the velocity of oscillations in time in m/s^2.

The rate of increase in the accelerations of oscillations equal to the third derivative of motion in time or the first derivative of acceleration of oscillations in time in m/s^3.

It has been established experimentally that all motor vehicle oscillations can be subdivided into *high-frequency* (5-13 Hz) and *low-frequency* (0.8-2 Hz) oscillations. Unsprung masses usually oscillate with a high frequency and sprung masses (body), with a low frequency.

High-frequency oscillations occurring even at low amplitudes (jolting, vibration) cause unpleasant sensations, but low-frequency oscillations are likewise unpleasant, since they tend to make one carsick. A man does not feel any oscillations in walking because he is accustomed to their frequency lying within 1.17-1.66 Hz from childhood. It should be noted that modern car bodies oscillate within approximately the same limits (1-1.3 Hz).

Changes in the oscillation frequencies affect the human organism more appreciably than changes in amplitude. Therefore, when strongly perceptible oscillations are estimated, the exponent of the frequency of oscillations is larger than that of their amplitude. Thus, a joint effect of the amplitude z_{max} and the frequency of oscillations n is described by a parameter equal to the product $z_{max}n^k$. Depending on the intensity of oscillations, the value of the exponent k is within 1.5-2.7. With the exponent $k = 2.7$ and the amplitude $z_{max} = 0.0065$ m the permissible value of the parameter is obtained at a frequency of $n = 3$ Hz.

When the oscillation rate increases the smoothness of run is worsened. The characteristic of oscillations versus their rate (in m/s) is illustrated below:

Nonperceptible	0.035
Hardly perceptible	0.035-0.1
Fully perceptible	0.1-0.2
Strongly perceptible . . .	0.2-0.3
Unpleasant and very unpleasant	0.3-0.4

The effect of alternating accelerations on the human organism largerly depends on the oscillation frequencies. When the frequency increases even slight oscillation accelerations may cause unpleasant or morbid sensations (Table 16.1).

TABLE 16.1

Accelerations of Oscillations

Frequency of oscillations, Hz	Accelerations in m/s² causing sensations	
	unpleasant	morbid
1	2.3	2.7
1.5	2.1	2.5
2	1.9	2.3
3	1.7	2.0

At the frequencies with which the vehicle body oscillates, the greatest effect is exerted on smooth running by the rate of change in the accelerations. Uneasy sensations appear at 25 m/s³ and unpleasant sensations, at 40 m/s³. In this way, the rate of increase in accelerations should not exceed 25 m/s³.

Table 16.2 illustrates the acceleration values permitted for healthy men of middle age.

TABLE 16.2

Permissible Values of Acceleration (in m/s²) for Various Kinds of Oscillations

Conditions	Oscillations		
	vertical	longitudinal	cross
Slow walking	1.0	0.6	0.5
Easy riding	2.5	1.0	0.7
Short-time travel	4.0	2.0	1.0

The German society of engineers has suggested a standard according to which the acceleration with oscillations of high amplitude and a frequency of 0.5-5 Hz and the rate with oscillations of low amplitude and a frequency of 15-80 Hz serve as the index used to estimate sensations.

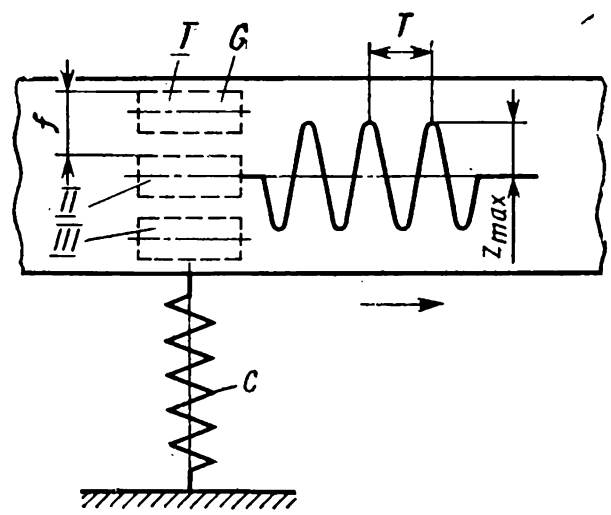


Fig. 16.2. Oscillation of a body with one degree of freedom

Let us establish the relationship between separate characteristics of the smoothness of run of a vehicle. With this aim in view, let us consider harmonic oscillations of a body with weight G (Fig. 16.2) and with one degree of freedom located on a spring of stiffness c (N/m).

If the spring is in a free state the body occupies position *I*. When the system is in equilibrium the spring is statically deformed under the action of the force G due to which the body takes position *II* and

$$f = \frac{G}{c} \tag{16.2}$$

Let the body be brought out of the state of equilibrium by shifting it to position *III* and compressing the spring and then letting the body oscillate. If the motion of this body is recorded on a paper tape moving at a constant speed as shown in Fig. 16.2 the recording will show the body oscillations. Let us mark on this curve the amplitude z_{max} equal to the maximum deflection of the body from the equilibrium position and the period T equal to the time interval which corresponds to the two nearest identical body positions (for example, the extreme upper ones).

During oscillations, the body moves nonuniformly, its position, velocity, acceleration and increase in acceleration changing at the same time. The nature of the curves of motion, velocity and accelerations is about the same.

The differential equation of oscillatory motion for this case has the form

$$m\frac{d^2z}{dt^2} + cz = 0$$

Solving this equation, we get

$$z = z_{max} \sin \sqrt{\frac{c}{m}}\, T \tag{16.3}$$

where z_{max} is the maximum amplitude in m.

The value of $\sqrt{\frac{c}{m}}$ in equation (16.3) is the angular frequency of free oscillations ω. Hence, equation (16.3) may also be written as

$$z = z_{max} \sin \omega t$$

Let us now find from the latter equation the dependences relating the individual characteristics of smooth running in the case of harmonic oscillations:

oscillatory velocity (in m/s)

$$v = \frac{dz}{dt} = z_{max}\omega \cos \omega t \tag{16.4}$$

acceleration of oscillations (in m/s^2)

$$j = \frac{d^2z}{dt^2} = -z_{max}\omega^2 \sin \omega t \tag{16.5}$$

rate of increase in accelerations (in m/s^3)

$$j' = \frac{d^3z}{dt^3} = -z_{max}\omega^3 \cos \omega t$$

The frequency ω and the frequency of oscillations n are mutually related by the equality:

$$n = \frac{1}{T} = \frac{\omega}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{c}{m}}$$

Inserting the value of stiffness c from (16.2) into this formula and expressing the mass m through G/g we can obtain

$$n=\frac{1}{2\pi}\sqrt{\frac{g}{f}}$$

Thus, the larger the static deformation (deflection) of the suspension, the less is the frequency of the natural oscillations. For this reason, soft suspensions are employed to diminish the frequencies of natural oscillations of the body and improve the riding comfort.

16.4. Oscillations of a Motor Vehicle

A simplified diagram of the oscillatory system of a motor vehicle is illustrated in Fig. 16.3*a* where the letters c_1 and c_2 denote the effective rigidities of the front and rear suspensions.

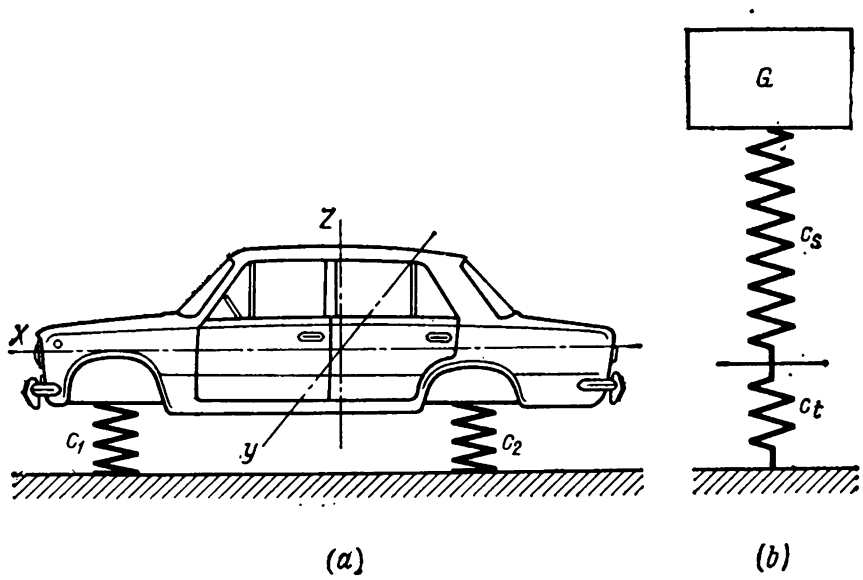


Fig. 16.3. Simplified diagram of the car oscillatory system and determining the effective rigidity of suspension:
(*a*) simplified diagram; (*b*) diagram for determining the rigidity of suspension

The effective rigidity of a suspension is the rigidity of an elastic element whose deflection is equal to the total deflection of the suspension and the tyres taking the same load. To determine the effective rigidity, Fig. 16.3*b* shows a weight G mounted on an elastic element of the suspension and the tyre the rigidities of which are respectively equal to c_s and c_t.

The weight G deforms the elastic system by an amount equal to the sum of deflections of the elastic element of the suspension and the tyre. At the same time the total deflection of this system

which is determined by the change in the position of the wheel axle (in m) is given by

$$f = \frac{G}{c_{eff}}$$

where c_{eff} is the effective rigidity of the suspension and the tyre in N/m.

Then,

$$\frac{G}{c_{eff}} = \frac{G}{c_s} + \frac{G}{c_t}$$

Solving the obtained equality with respect to the rigidity c_{eff} (in N/m), we get

$$c_{eff} = \frac{c_s c_t}{c_s + c_t} \tag{16.7}$$

The rigidity of the front or rear suspension of modern motor vehicles is within 20-60 kN/m and that of tyres, within 200-450 kN/m. The smaller values of rigidity refer to cars and larger values, to trucks.

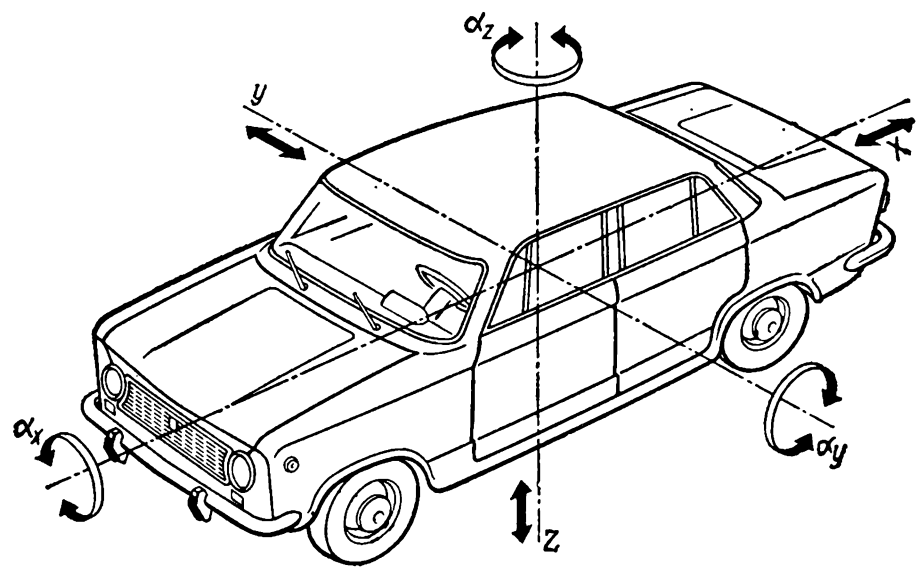


Fig. 16.4. Kinds of body oscillations

The body of a vehicle has six degrees of freedom (Fig. 16.4) and can perform six different oscillations. The linear motions along axes *xx*, *yy* and *zz* are denoted respectively as S_x, S_y and S_z and the angular motions with respect to these axes, as α_x, α_y and α_z.

Ordinarily, due to the complexity of investigating a system with six degrees of freedom and in order to simplify the calculation of the suspension, the body of a vehicle is regarded as a system with two degrees of freedom, i.e., account is taken only of *jumping* S_z and *galloping* α_y. These oscillations are of prime impor-

tance for comfort because they produce the most morbid sensations in the human organism.

Vertical oscillations of the body appear when the wheels go over irregularities. If a vehicle is mounted on a soft suspension, slowly damped vertical oscillations of the body occur in this case due to high deformations of the elastic elements of the suspension. Sometimes these oscillations are intensified, which can be attributed to resonance appearing when the frequency of application of the perturbing force is close to one of the frequencies of the body natural oscillations.

Soft suspension and shock absorbers are utilized to reduce vertical body oscillations causing unplesant sensations.

Let us get acquainted with the basic principles of the theory of oscillations but first and foremost with the elasticity centre of the system to see how α_y can be reduced.

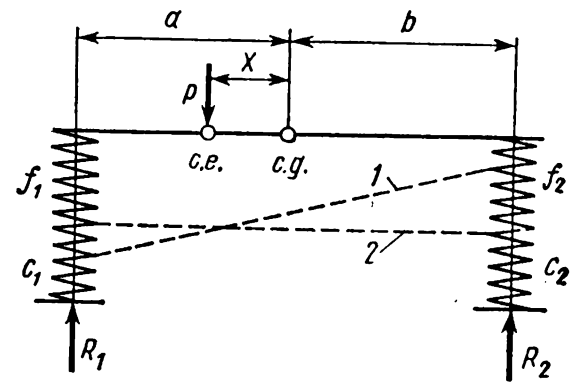


Fig. 16.5. Determining the position of the elasticity centre

The elasticity centre of a system is the point where only linear shift of the system takes place as a result of the application of an external perturbing force. In order to determine the position of the elasticity centre, we consider a bar resting on elastic elements as shown in Fig. 16.5.

If the external perturbing force P is applied not to the centre of elasticity (c. e) both linear and angular shifts of the bar occur (position *1*). When the force P is applied to the centre of elasticity only a linear shift takes place (position *2*). In the latter case the deflections of the suspensions f_1 and f_2 are mutually equal and α_y is absent.

Let us determine x, i.e., the distance from the centre of elasticity to the centre of gravity (c. g). It follows from the equilibrium condition of the bar that

$$\Sigma M_{c.g} = R_1 a - Px - R_2 b = 0$$

where R_1 and R_2 are bearing reactions.

Solving this equation with respect to x, we get

$$x = \frac{R_1 a - R_2 b}{P}$$

where x is expressed in m

Replacing the reactions R_1 and R_2 (in N) by the products of rigidity for bending we obtain

$$R_1 = c_1 f_1 \text{ and } R_2 = c_2 f_2$$

where c_1 and c_2 are the rigidities of the elastic elements.

In this case

$$P = R_1 + R_2 = c_1 f_1 + c_2 f_2$$

Inserting the found values of R_1, R_2 and P into the expression for determining the distance x, we get

$$x = \frac{c_1 f_1 a - c_2 f_2 b}{c_1 f_1 + c_2 f_2}$$

But since, according to the condition,

$$f_1 = f_2$$

then

$$x = \frac{c_1 a - c_2 b}{c_1 + c_2} \tag{16.7}$$

As a rule, the body of a vehicle oscillates with a frequency of free oscillations. In order to study free oscillations of a vehicle in the longitudinal vertical plane, the sprung mass, i.e., the mass m_b of the body (Fig. 16.6*a*) is replaced by three masses: m_1, m_2 and m_3 linked by a weightless bar (Fig. 16.6*b*). The masses m_1 and m_2 are positioned at distances a_b and b_b from the centre of gravity of the body and the mass m_3, at its centre of gravity. For a three-mass system to correspond dynamically to the actual mass of the sprung portion of the vehicle, the following conditions must be observed:

the sum of all masses in the system should be equal to the sprung mass of the vehicle

$$m_1 + m_2 + m_3 = m_b \tag{16.8}$$

the centre of gravity of the system should coincide with the centre of gravity of the sprung mass of the vehicle, i.e.,

$$m_1 a_b = m_2 b_b \tag{16.9}$$

the moment of inertia of the system with respect to the horizontal axis *yy* passing through the centre of gravity and perpendicular to the plane, where the masses of the system are, should be equal to the moment of inertia of the sprung mass with respect to the same axis, i.e.,

$$m_1 a_b^2 + m_2 b_b^2 = \mathrm{I} = m_b \rho_b^2 \tag{16.10}$$

where ρ_b is the radius of inertia of the sprung mass of the vehicle with respect to the axis *yy* (see Fig. 16.4).

Solving equations (16.8)—(16.10), we find the m_1, m_2 and m_3 masses (in kg):

$$m_1 = \frac{m_b \rho_b^2}{a_b L} \tag{16.11}$$

$$m_2 = \frac{m_b \rho_b^2}{b_b L} \tag{16.12}$$

$$m_3 = m_b \left(1 - \frac{\rho_b^2}{a_b b_b}\right) \tag{16.13}$$

If the bar is moved out of the equilibrium position and then released it will begin to oscillate on the elastic elements. Let us

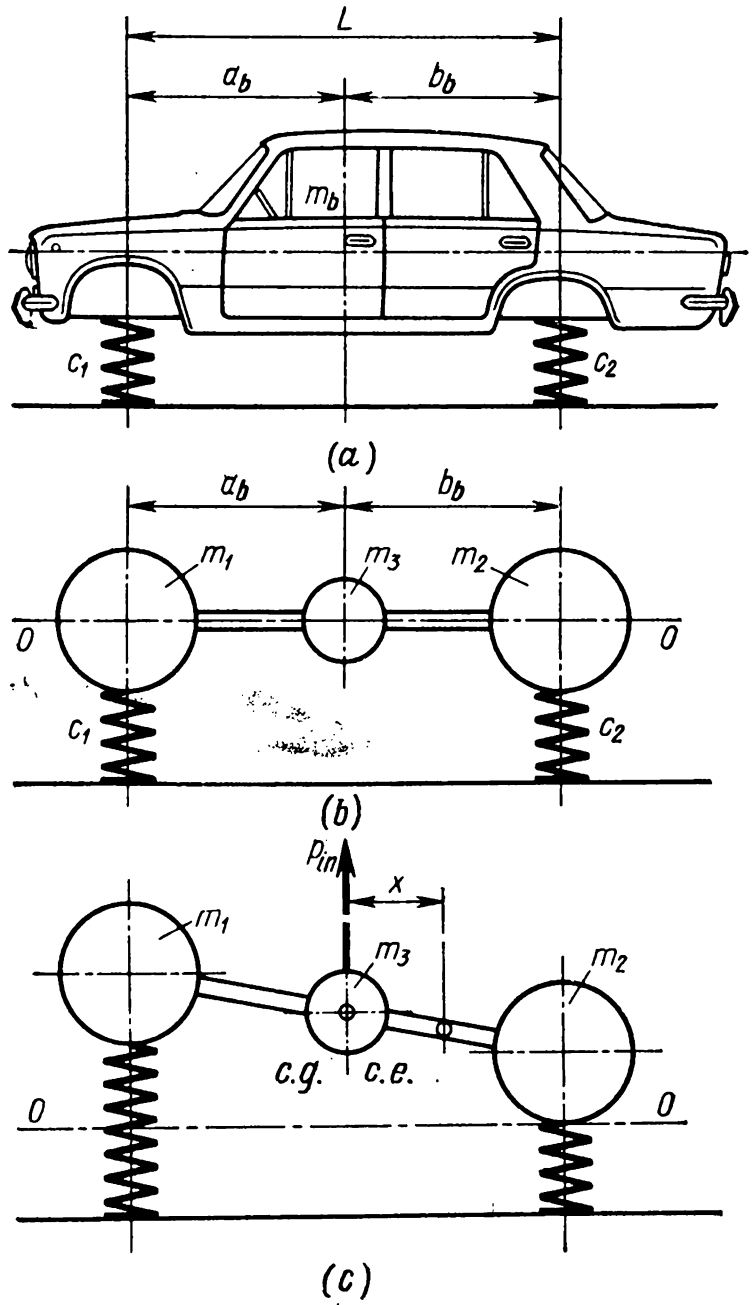


Fig. 16.6. Free oscillations of a body:
(*a*) diagram of body suspension; (*b*) system equivalent to sprung mass; (*c*) of the centre of gravity and the centre of elasticity

consider the effect exerted by the mass m_3 and the elasticity centre position on the oscillations of the bar (Fig. 16.6*c*). During oscillations, the force of inertia applied to the centre of gravity of the system (in N) appears:

$$P_{in} = m_3 j$$

The force of inertia creates a moment relative to the elasticity centre (in N·m)

$$M_{in} = P_{in} x = m_3 j x$$

To prevent further increases in the oscillations of the system. the moment M_{in} should be as small as possible or equal to zero. The moment M_{in} will be equal to zero if the mass $m_3 = 0$ or the arm x of the force P_{in} is likewise equal to zero.

Let us consider a case when the mass m_3 is equal to zero. It follows from equation (16.13) that the mass $m_3 = 0$, when $\frac{\rho_b^2}{a_b a_b} = 1$, because the body mass m_b cannot be equal to zero. To obtain this equality, the designer either correspondingly distributes the masses of the vehicle by repositioning, for example, the engine further forward and the luggage carrier backward ensuring thereby the required value of ρ_b, or if possible he may change the position of the centre of gravity. For modern cars the ratio $\frac{\rho_b^2}{a_b b_b}$ is close to unity, in view of which they have a good smoothness of run.

Let us consider another case when the arm $x = 0$, i.e., when the centre of gravityy coincides with the centre of elasticity. In this case

$$x = \frac{c_1 a_b - c_2 b_b}{c_1 + c_2} = 0$$

and therefore

$$c_1 a_b = c_2 b_b \text{ or } \frac{c_1}{c_2} = \frac{b_b}{a_b}$$

Hence, the rigidities of the suspensions should be selected so that they are inversely proportional to the centre of gravity coordinates. Then, if the front and rear suspensions are deflected by the same amount the body of the vehicle will move vertically without galloping.

The values of the vertical motion S_z are approximately determinated after specifying the frequency of oscillations n within recommended limits (1.1-1.3 Hz).

The rocking α_x (see Fig. 16.4) is caused by the action of impact forces taken by the wheels on one side of the vehicle when it comes against a single irregularity. In this case moments

which generate the transverse body oscillations appear and disappear. Rocking is decreased by using a transverse stabilizer which improves the angular rigidity of the suspension.

Apart from oscillations due to road irregularities a vehicle is also subjected to high-frequency vibrations with low amplitudes generated by the engine, transmission and tyres. Vibrations exert practically no effect on the smoothness of run but may produce noise and destroy the joints between parts and units. Vibrations can be eliminated or lessened by using vibroabsorbing gaskets and special coverings, by better quality of manufacture, assembly and balancing of the parts to prevent resonance. Seats with backs of special design are employed to reduce the effect of oscillations and vibrations felt by the driver and the passengers.

16.5. Effect of Design Factors on Smooth Running

Effect of Tyres

Pneumatic tyres supplement the elastic elements of the suspension and exert an appreciable effect on the smoothness of run of the motor vehicle. Being highly elastic, a tyre diminishes or damps the noise produced in motion on a rough surface and "absorbs" fine road irregularities because of tread deformation. The less the air pressure in the tyre, the more the tread deformed and the smaller the vertical motions of the wheel centre.

The tyre negligibly affects low-frequency oscillations, since its deformation under a static load is small (only 15-35 mm) as compared with a static deflection of the suspension required for smooth running (100-250 mm). On the other hand, the tyre essentially affects high-frequency oscillations because a drop in the rigidity of tyres reduces the vertical movement of the wheels and accelerations of the body. It is therefore better to use tyres having the least possible rigidity to improve the smoothness of run. For this reason the tendency in the development of designs of modern tyres is to reduce their stiffness by reducing their air pressure and increasing their profile width.

However, it is a mistake to think that soft tyres allow the designer to dispense with elastic suspension elements. This can be explained by the fact that to reduce losses due to rolling and heating, everything is done to ensure the minimum tyre internal friction whose decrease would, in turn, result in a very slow attenuation of the body oscillations without the elastic suspension elements. Besides, road roughness would maintain the initiated oscillations. Therefore, on average, the body oscillations would be more intensive without elastic suspension elements than with them, all other conditions being equal.

Advantages of Independent Suspension

As regard smooth running, independent suspension offers a number of advantages compared with the conventional suspension. At present independent suspension is mainly utilized for the front wheels of cars. For all that, some cars and special vehicles are designed with independent suspension of the front and rear wheels. Whereas in the case of the conventional suspension the right and left wheels are linked rigidly by a beam or the axle housing, each wheel with the independent suspension is hinge-jointed with the body or frame by means of one or two arms. With the conventional suspension, when one of the wheels runs against an irregularity, the right and left wheels are tilted in a transverse plane through the same angle, which causes the body to roll. In the case of the independent suspension vertical motion or lateral tilt of one of the wheels do not involve the other wheel and the body is hardly inclined at all in the transverse plane.

Independent suspension for the front wheels allows larger static deflections to be obtained, which, in turn, ensures a static deflection ratio of the front and rear suspensions close to unity $\left(\frac{f_{fr\ st}}{f_{rear\ st}} \approx 1\right)$. As a result, when a vehicle runs against an obstacle there is practically no galloping because the body moves parallel to itself.

Since the beam of the front axle and leaf springs are absent in independent suspension, the weight of unsprung parts of the vehicle is reduced and the frequency of natural oscillations of its body is increased. It has been found from tests that an increase in the natural oscillation frequency reduces the length of irregularities which cause resonance oscillations of the wheels. The shorter the irregularity, the smaller on average is its height and the intensity of oscillations it causes. Therefore, it is expedient to decrease the mass of the unsprung parts. This also reduces the load on the shock absorbers and improves the conditions for damping wheel oscillations.

It should be noted that the use of independent suspension for the front wheels does not noticeably increase the vehicle cost. When independent suspension is employed for the rear driving wheels the vehicle cost increases because this system considerably complicates the design of the wheel drives. For this reason, independent suspension is, as a rule, utilized only for the front wheels. When used for the rear driving wheels, it also slightly improves the smoothness of run. Today, there are no sufficiently well-grounded reasons for the more complicated and costly independent suspension of the rear driving wheels.

Thus, independent suspension for the front wheels is rational.

For its elastic element, springs, more rarely torsion bars and lately pneumatic elastic elements are used. The rear suspension is normally a conventional one with leaf springs. Springs and torsion bars are superior to leaf springs in their smaller mass, longer service life; they operate practically without friction, are simple in manufacture and require no maintenance. Torsion bars last for a shorter time and they are more complicated to make than ordinary springs. At the same time torsion bars distribute loads more favourably between the frame elements and reduce the mass of unsprung parts because some of the mass of the torsion bars is included in the sprung mass of the vehicle.

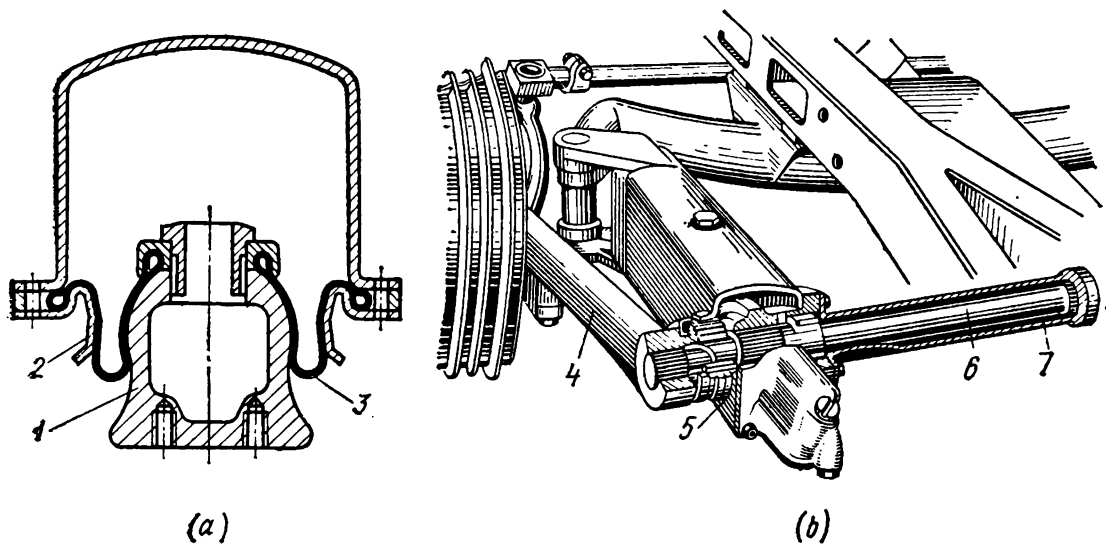


Fig. 16.7. Independent suspensions
(*a*) diagram of a pneumatic elastic element; (*b*) torsional suspension; *1*—piston group; *2*—housing; *3*—diaphragm; *4*—arm; *5*—steering knuckle; *6*—rod-type torsion bar; *7*—tubular torsion bar

Pneumatic suspension is composed of systems intended to compress the air and regulate the elastic characteristic of the suspension as well as of elastic elements and auxiliary devices. Ordinarily pneumatic suspension is designed according to the following scheme. A compressor delivering air into a reservoir under a pressure of 0.6-1.2 MN/m^2 is mounted on the vehicle. The reservoir admits compressed air to the elastic elements (Fig. 16.7*a*) representing housings *2* linked with the sprung mass. The housings contain piston group *1* linked with the unsprung mass. The housing and the piston group are connected by diaphragm *3* for better sealing and to lessen friction between the members of the elastic elements. The wheel as it hits an irregularity is raised, the air is additionally compressed, and this softens the shock taken by the body. A special regulator ensures the same distance between the wheel and the vehicle frame under various loads.

Among the advantages offered by the pneumatic suspension are: high smoothness of run due to the low rigidity and favour-

able nature of change in the elastic characteristic; regulation of the suspension rigidity within wide limits; keeping the body at a permanent height under all static loads taken by the elastic elements; possibility of changing the body height; the long service life of the elastic elements; small mass of their unsprung parts; high response of the suspension to roughness due to the almost complete absence of friction in the elastic elements; and a longer service life of the vehicle due to improved smoothness of run. However, the pneumatic suspension is intricate in design and costly. For this reason it is employed at present only when the mass of the sprung parts changes within wide limits (rear suspension of trucks and buses, suspensions of trailers) or when the smoothness of run imposes on the vehicle special requirements caused by the need to change the elastic characteristic of the suspension.

Torsional suspension has a metal torsion bar operating in torsion. Torsion bars can be installed both crosswise and along the axis of the vehicle. Depending on their design, all torsion bars can be classified as *single* in the form of a solid round bar or less frequently of tubular cross section, as *bunched* usually consisting of one central and six peripheral bars, and *leaf-spring* (several thin bands). Leaf-spring and bunched torsion bars are utilized when the motor vehicle layout requires shorter torsion bars. But their mass as compared with a single torsion bar, is 1.5-1.8 times larger. In some torsional suspension designs the torsion is regulated by pretwisting the bar. Sometimes various types of torsion bar combinations can be employed.

Figure 16.7*b* shows a front one-arm independent suspension with a king pin fixed to the frame. The suspension has a combination-type torsion bar, i.e., a tubular torsion bar *7* accommodating a rod-type torsion bar *6*. One end of torsion bar *6* is connected by splines with arm *4* rigidly linked with the wheel or the axle of the vehicle and the other, with tubular torsion bar *7*. The other end of torsion bar *7* is rigidly connected to steering knuckle *5*. When the wheel moves the rod and the tube are twisted forming an elastic link between the wheel and the steering knuckle (frame). The length of a combination torsion bar is composed of the lengths of the tube and the rod to make the suspension softer.

The high internal friction in a leaf-spring torsion bar makes the work of the shock absorbers easier, and makes them unnecessary if these torsion bars are installed on light vehicles.

Chapter XVII TYPES OF TRANSMISSION

Transmissions may be *mechanical, hydromechanical* or *electromechanical.*

The most popular is the *mechanical transmission* composed of a clutch, gearbox, universal-joint drive, and driving axle, the mutual arrangement of which depends on the layout of the motor vehicle.

With the engine in front and a rear-wheel drive (Fig. 17.1*a*), a uniform load distribution can be assured between the axles, but a long universal-joint shaft is required. When the engine is in front and the vehicle has a front-wheel drive (Fig. 17.1*b*) a more compact transmission is obtained because the universal-joint drive is no longer necessary, and the clutch and the gearbox are combined with the driving axle.

However, the use of such transmission is limited because the dynamics of the vehicle on upgrade is impaired due to a fall in adhesion weight as a result of the load redistribution on the axle.

The engine rear arrangement (Fig. 17.1*c*) employed in some cars and buses allows a better view of the road and reliably separates the engine from the passengers, reducing the noise and penetration of exhaust gases into the saloon. The shortcoming of this layout is the complicated control of the engine, gearbox and clutch, since they are positioned far from the driver. Besides, with such layout the driver and the front passenger sit near the front of the vehicle and in the case of collision or striking an obstacle they may be severely injured.

A vast variety distinguishes mechanical transmissions of cross-country vehicles. In a double-axle vehicle with a drive to both axles (Fig. 17.2*a*) the transmission, besides clutch *1*, gearbox *2*, universal-joint drive *3* and rear driving axle *4* common for ordinary vehicles with a rear-wheel drive, also includes distributor box *5* (Fig. 17.2*a*) sometimes combined with an auxiliary transmission case, universal-joint drive *6*, and front driving axle *7*, which has hinge joints of equal angular velocities *8* in addition to the mechanisms common for the driving axle.

Three-axle motor vehicles with a drive to all axles are provided, apart from the above-enumerated units, with one more driving

axle *10* linked by universal-joint drive *9* with distributor box *5* (Fig. 17.2*c*) or with the middle driving axle (Fig. 17.2*b*).

Cross-country vehicles frequently carry an interaxle differential which distributes torque equally between the middle and rear driving axles thereby preventing their overload.

A mechanical transmission is highly reliable and is distinguished by a high efficiency, which makes it very popular. At the same time it has a number of shortcomings and this impels the

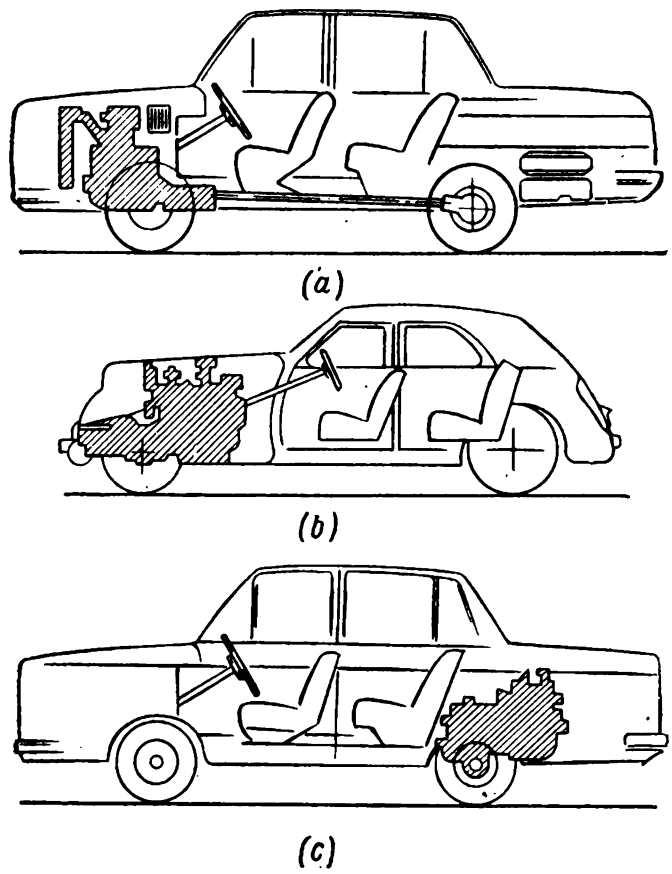


Fig. 17.1. Layout of mechanical transmission elements of cars
(*a*) engine in front with rear-wheel drive; (*b*) engine in front with front-wheel drive; (*c*) engine in rear with rear-wheel drive

designers developing transmissions to seek for solutions new in principle. For instance, when the gears are changed the flux of power from the engine to the driving wheels is interrupted. In this case the motion of the vehicle is retarded and the acceleration dynamics are impaired. The gear selection and the moment of its engagement depend on the skill of the driver and this does not allow the ability inherent in the vehicle design to be utilized to full advantage. The mechanical transmission of multiple-drive vehicles become cumbersome and their operation noisy. The linings of the driven disks in friction clutches incorporated in the mechanical transmission are rapidly worn out, and the clutch has to be frequently readjusted and repaired. These shortcomings can

in part be eliminated by combining a mechanical transmission with a hydraulic or electrical device.

The hydraulic units which operate on the principle of utilizing the kinetic energy of fluid make the operation of the transmission smooth and noiseless and damp torsional oscillations arising in it.

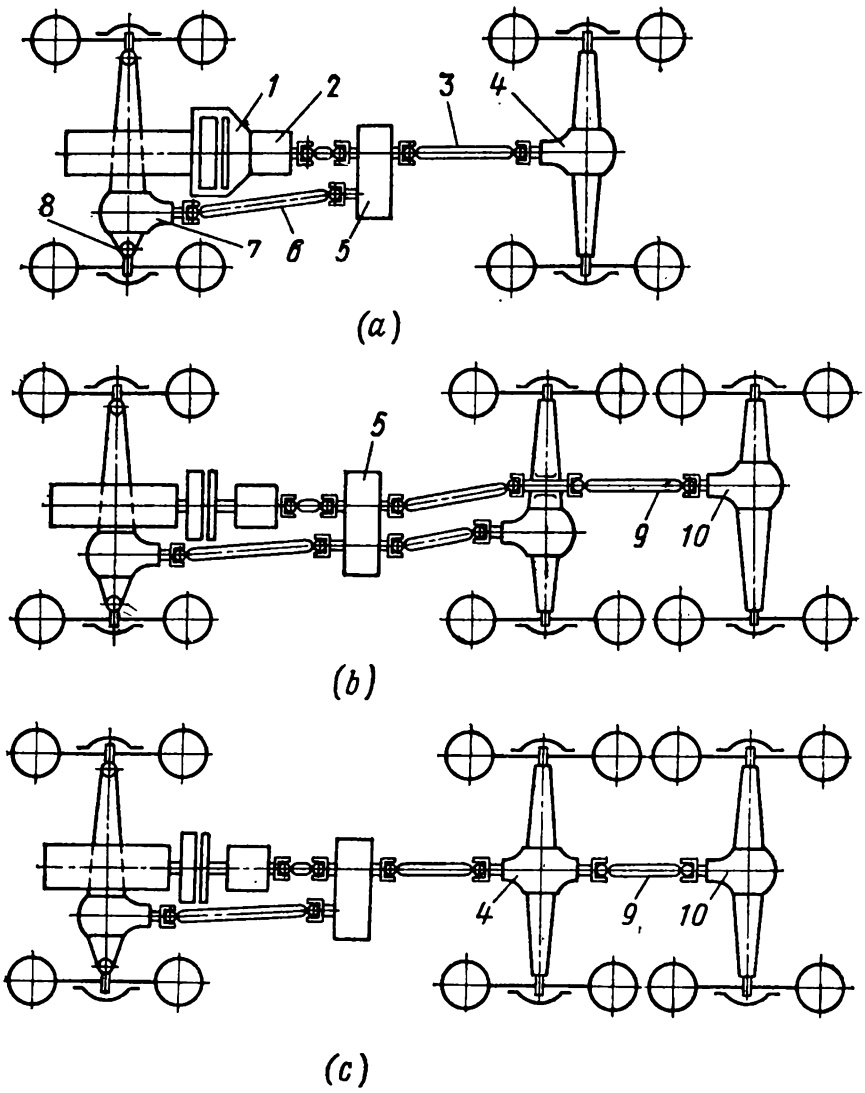


Fig. 17.2. Mechanical transmissions of cross-country trucks
(*a*) drive to both axles; (*b*) and (*c*) drive to three axles; *1*—clutch; *2*—gearbox; *3*, *6* and *9*—universal-joint drives; *4*—rear driving axle; *5*—distributor box; *7*—front driving axle; *8*—hinge joint of equal angular velocities; *10*—second rear driving axle

The *hydromechanical transmission* of a vehicle contains a hydraulic clutch or a hydrotransformer.

The *hydraulic clutch* smoothly transmits torque from the engine to the gearbox thus facilitating the operation of the friction clutch.

Let us discuss the operation principle of the hydraulic clutch. Let some fluid be poured into a vessel divided by partitions and set the vessel in rotation. The fluid will be thrown upwards and sideways (Fig. 17.3*a*). If the vessel is covered by a lid with iden-

tical partitions (Fig. 17.3*b*) the fluid striking against the partitions of the lid will entrain it with itself and then flow down in a uniform stream into the vessel. Thus, the hydraulic clutch does not possess any positive feedback between the driving and driven parts, which ensures smooth torque transfer.

The parts of an automobile hydraulic clutch are illustrated in Fig. 17.3 *c*. Pump *3* with radial blades is welded to front portion *2*

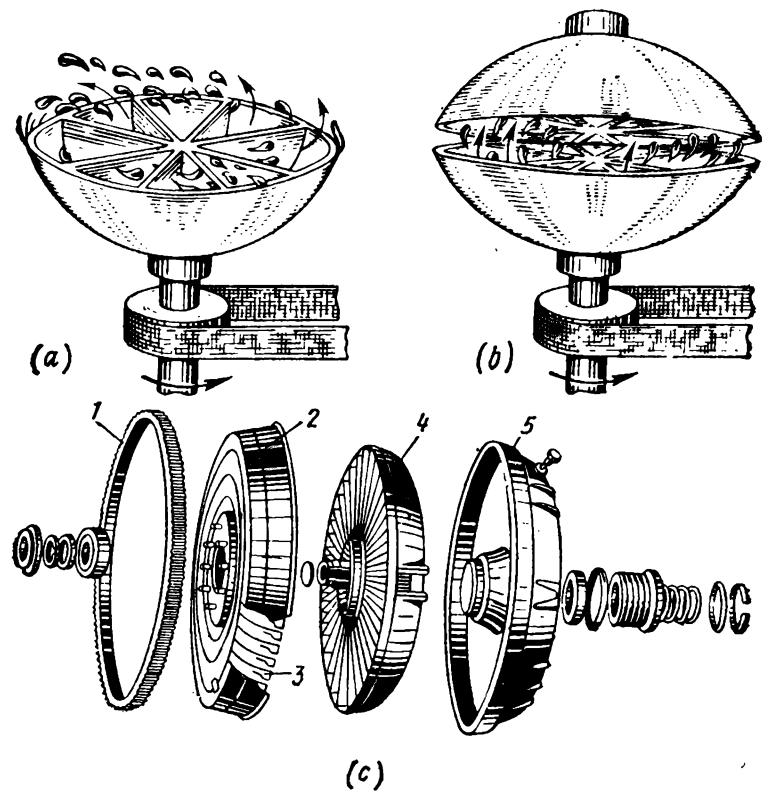


Fig. 17.3. Operating diagram of hydraulic clutch and its principal parts

(*a*) and (*b*) operating diagrams of hydraulic clutch; (*c*) principal parts of hydraulic clutch; *1*—toothed rim of flywheel; *2*—front portion of housing; *3*—pump; *4*—turbine; *5*—rear portion of housing

of the hydraulic clutch housing. During assembly, rear portion *5* of the housing is welded to the front one after installing turbine *4*, also having radial blades, into the housing. The housing is rigidly linked with the engine crankshaft and the turbine, with the primary shaft of the gearbox. Turbine oil of low viscosity is poured into the housing through the hole closed by a plug.

At a low angular velocity of the pump the kinetic energy of the fluid is not high and if the drive is engaged the turbine remains stationary. In this case the hydraulic clutch slips. As the angular velocity of the pumps increases the torque transmitted to the turbine grows in magnitude, the vehicle starts and smoothly picks up speed. Since the load on the engine increases gradually, acceleration from rest can be started in second gear. As the hydraulic

clutch slips, the vehicle can move at a low speed even in direct drive. This reduces the number of gear changes and makes steering easier. However, it is impossible to completely disconnect the engine from the transmission when using a hydraulic clutch because the turbine continues to rotate even at small angular velocities of the pump and with the drive disengaged. Thus, the hydraulic clutch does not ensure shockless gear changes and it is therefore usually mounted together with a friction clutch.

If during the motion the driver slightly opens the throttle without disengaging the drive the angular velocity of the turbine

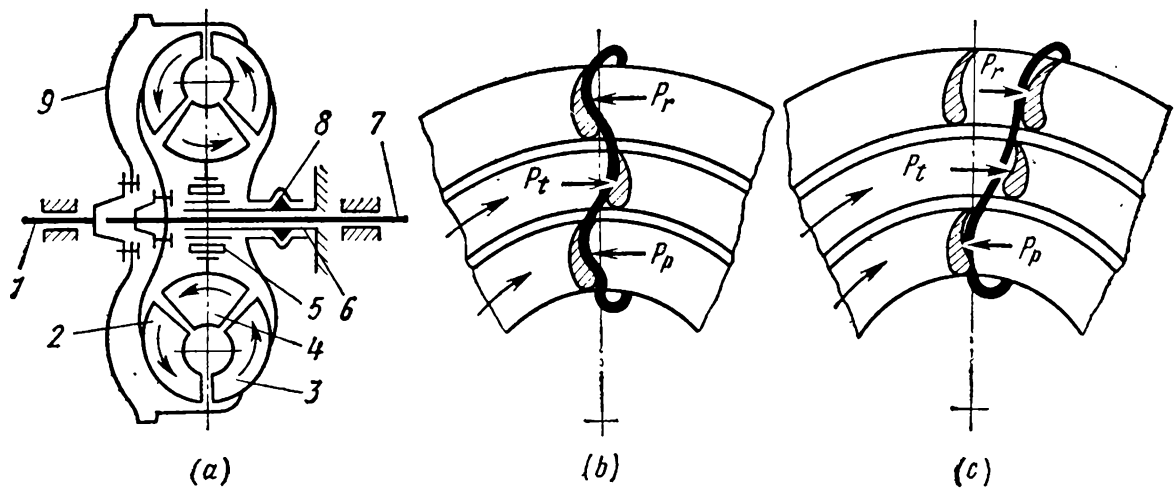


Fig. 17.4. Design diagrams of hydraulic transformer and its operation

(*a*) design diagram of hydraulic transformer; (*b*) and (*c*) operating diagrams of hydraulic transformer; *1*—crankshaft; *2*—turbine; *3*—pump; *4*—reactor; *5*—free-wheeling clutch; *6*—bushing; *7*—primary shaft of gearbox; *9*—gland; *9*—housing

will be higher than that of the pump. The fluid entrained by the turbine strikes against the pump vanes and begins to rotate the crankshaft at a high rate. The braking torque applied at this moment to the driving wheels will stop the running vehicle much faster. For all that the engine of a vehicle with a hydraulic clutch will be braked more effectively if the vehicle moves at a speed higher than 7-8 m/s. This is a disadvantage of the hydraulic clutch.

The hydraulic transformer changes smoothly and automatically the transmission gear ratio making it unnecessary to shift the gears. It is designed on the same principle as the hydraulic clutch but differs from the latter by the number of working wheels and the curvilinear shape of the blades. The diagram of a hydraulic transformer is illustrated in Fig. 17.4*a*. Crankshaft *1* of the engine is rigidly linked with housing *9* and pump *3*. Turbine *2* is connected with primary shaft *7* of the gearbox. Reactor *4* is arranged between the pump and the turbine. The inner space of the hyd-

raulic transformer is filled with low-viscosity oil whose leakage is prevented by gland *8*.

As the crankshaft rotates, the oil caught by the pump is thrown to the outer edge of the pump wheel under the action of the centrifugal force, strikes the blades of the turbine and drives it. The oil from the turbine enters reactor *4* changing the direction of the fluid flow and then goes again into the pump.

If the turbine rotates slowly the fluid admitted into the reactor strikes its blades from inside and tends to turn the reactor in the direction opposite to the pump and turbine rotation with the force P_r (Fig. 17.4*b*). In this case the force P_r is directed in the same way as the force P_p acting on the pump and oppositely as compared with the force P_t which acts on the turbine. The moment of the force P_t applied to the turbine is equal to the sum of the moments of the forces P_p and P_r. Thus, the moment on the primary shaft of the gearbox proves higher than the moment of the engine. As the resistance to the motion of the vehicle increases, the angular velocity of the turbine reduces. The pressure exerted by the fluid on the reactor grows and automatically increases the turbine torque and hence the tractive force on the vehicle driving wheels.

With an increase in the angular velocity of the turbine due to the reduction of engine load, the fluid coming from the turbine strikes the blades of the reactor from outside (Fig. 17.4*c*). The force P_r changes its direction and, as a result, the torque on the turbine becomes equal to the difference of the moments of the forces P_p and P_t. Hence, the torque on the primary shaft proves less than the engine torque. In order to avoid an excessive drop of the moment, reactor *4* is coupled with stationary bushing *6* by means of free-wheeling clutch *5*. If the angular velocity of the turbine is smaller than that of the pump, clutch *5* jams the reactor on the bushing and the reactor remains stationary (does not rotate). After the angular velocities of the turbine and the pump are equalized the clutch unwedges the reactor and the latter begins to rotate in the same direction as the turbine. In this case the blades of the reactor do not change the fluid flow direction, the moment is no longer reduced and the hydraulic transformer operates as the hydraulic clutch.

The shortcomings of a hydraulic transformer are a low efficiency, a large mass and impossibility to increase the torque more than 2.5-3 times. To increase the torque, the design of the hydraulic transformer has to be made much more complicated and a stepped gearbox installed.

Electric energy is utilized to drive the *electromechanical transmission*. The torque in an *electromagnetic clutch* is transmitted from the driving to the driven parts through fine iron powder

contained between them. The clutch mounted on flywheel *1* (Fig. 17.5) has driving element *2* with field winding *3* and driven element *4* connected to primary shaft *5* of the gearbox. The driving and driven elements are separated by a radial gap whose size does not change when the clutch operates. The gap space is filled with fine iron powder.

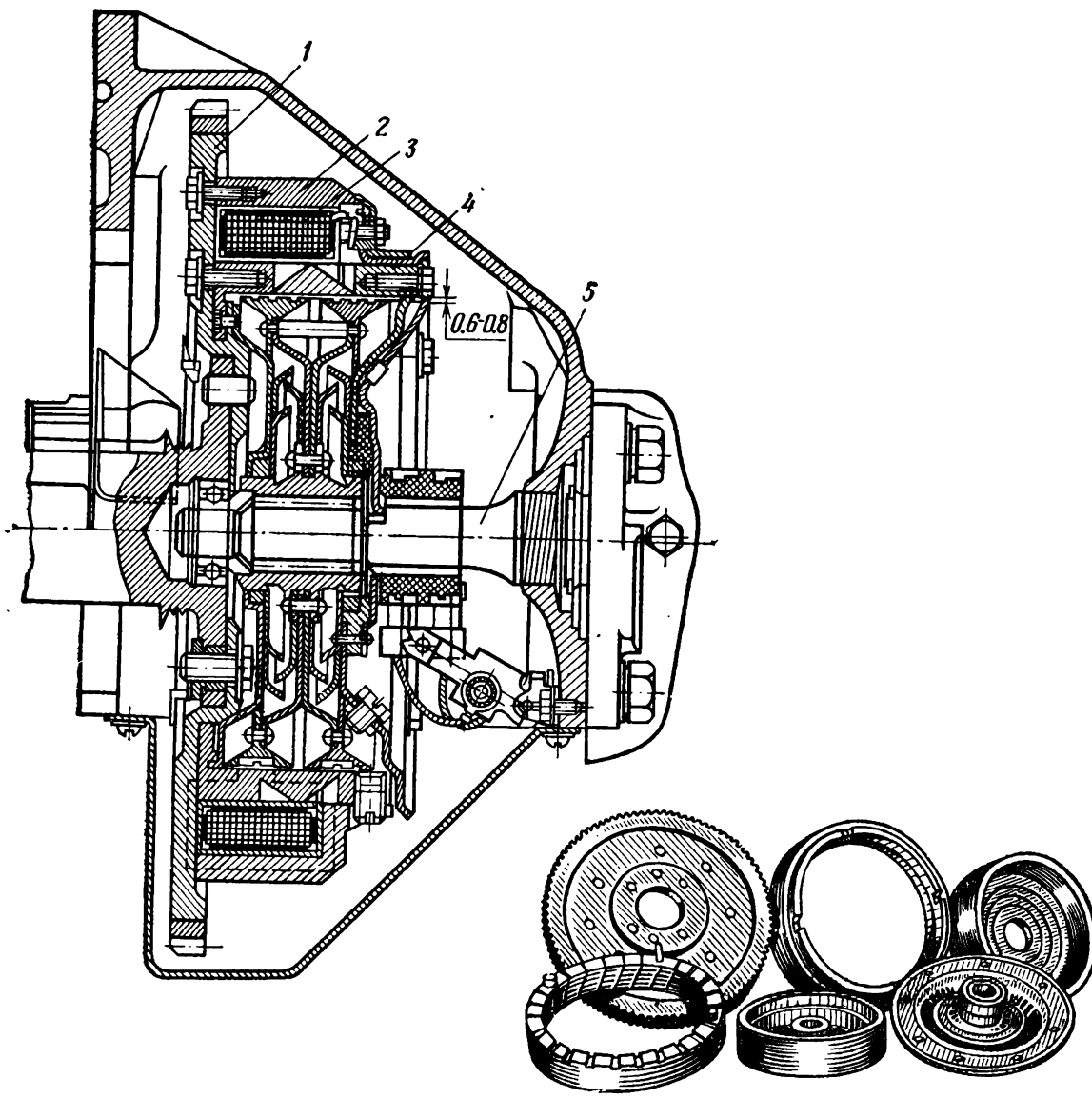


Fig. 17.5. Electromagnetic powder clutch
1—flywheel; *2*—driving element; *3*—field windings; *4*—driven element; *5*—primary shaft

If current does not pass through the field winding the driven element is not linked with the driving element because the powder is forced to the surface of the driving element by centrifugal forces and a slight gap appears between the powder and the driven element. If the field winding is connected to a current source the magnetic field built up by the current will magnetize the powder particles which will be attracted one to another and also to the working surfaces of the driving and driven elements to connect them. The torque transmitted by the clutch depends on the force

with which the powder particles are attracted, and on the friction coefficient between them and the clutch working surfaces.

Thus, the force of the pressure springs is replaced by the electromagnetic forces in the clutch being considered and the role of the friction surfaces is played by powder particles. As the vehicle is started from rest the current passing through the field winding is reduced and the clutch slips. The electromagnetic powder clutch is practically wearless, since even a prolonged slipping only shifts the particles of the powder and the reduction in their size has almost no effect on the operation of the clutch.

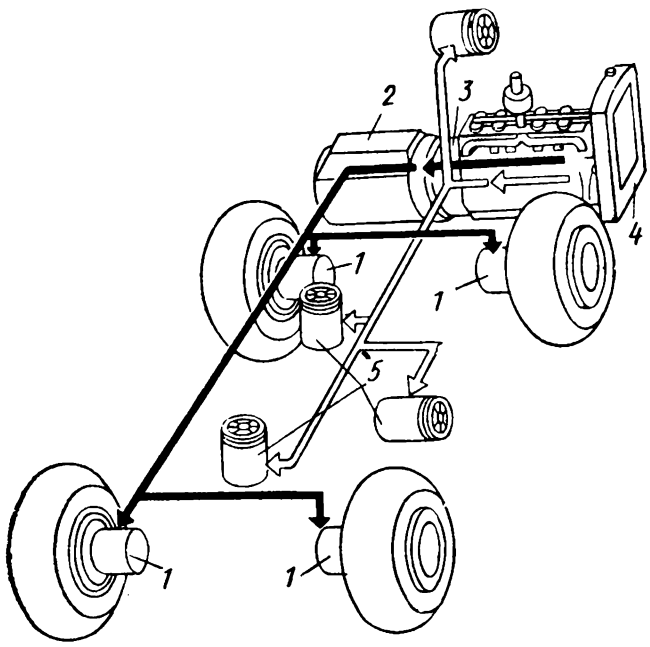


Fig. 17.6. Motor vehicle with an electromechanical transmission

1—electric traction motors; *2*—*d-c* generator; *3*—*a-c* generator; *4*—diesel; *5*—electric motors of auxiliary mechanisms

The use of electric energy in a transmission makes it possible to use trailer trains of any length, since there is no need for a mechanical coupling between the engine and the driving wheels. The energy is converted twice in such a transmission: the mechanical energy of the engine is first converted into electric energy by a generator which drives the electric traction motors thus converting it again into mechanical energy.

By way of example, Fig. 17.6 shows a vehicle with an electric drive to all wheels. The source of energy is diesel *4* rotating the armature of d-c generator *2* and the rotor of a-c generator *3*. Generator *2* feeds electric traction motors *1*. Electric motors *1* with gear reducers are built into the driving wheels called motor-wheels. These wheels appreciably improve the vehicle motion because skidding of one of the wheels does not affect the operation of the others. Generator *3* supplies power to electric motors *5* of the auxiliary mechanisms.

When braking the vehicle the electric traction motors act as generators. In this case the electric motors do not absorb but produce electric energy which is converted into heat in load resistors. With electric braking, the wheels of all trailers are braked simultaneously. This eliminates skidding and does not allow the rear trailer to run against the front ones.

Among the shortcomings of an electromechanical transmission are its large mass and low efficiency.

Chapter XVIII FUTURE DEVELOPMENTS OF MOTOR VEHICLE DESIGNS

The number of vehicles in the world is incessantly increasing. The annual average increase during the last ten years is to about six million vehicles. The total number of vehicles registered on January 1, 1974 comprises nearly 285 million, 75 per cent cars, about 24 per cent trucks and approximately 1 per cent buses.

The quality of vehicles is improved alongside with their number. Work is conducted in many countries to reduce the mass of motor vehicles by a better utilization of the properties of metals and the use of light alloys and plastics. The casings of gearboxes and clutches, frames, panels of bodies and cabs are preferably made of aluminium alloys. Plastics are extremely popular. They, as compared with metals, have a lower specific weight, are more plastic, a fact very important for the manufacture of intricate shaped parts, and effectively resist corrosion.

Trucks designed in the recent years have the driver's cabin mounted above the engine. This makes it possible for the length of the load platform to be increased, without changing the base of the vehicle, and its load-carrying capacity improved by making the front axle bear greater loads. For the same purpose high-capacity trucks and buses sometimes have the engine installed between the front and rear axles, under the floor of the body.

The general trend in the development of motor vehicle engines and their future designs has been considered above in Chapter VI.

When designing the transmission of a vehicle much attention is given to the automation of gearbox control on cars and trucks of low and large load-carrying capacity. The towing truck intended for intercity transportation, which have to move at highest speeds and the least consumption of fuel, are equipped with semiautomatic clutches and also 10- and 12-step gearboxes which offer the choice of the drive most advantageous as to the conditions of motion. Large-capacity trucks and buses employ gearboxes with pneumatic boosters to change the gears.

Wide use is also made of automatic hydraulic drives which improve the dynamics and make the control of a vehicle easier especially in urban conditions. If a vehicle is provided with a hydraulic drive the driver starting his vehicle from rest presses the button for forward run and then manipulates only the throttle pedal. A hydraulic drive also makes movement safer since the dri-

ver uses two pedals only (the throttle and brake) instead of three (clutch, throttle and brake control) as in ordinary motor vehicles.

For all that, as compared with a mechanical gearbox, the hydraulic drive is more complicated in design, has a larger mass and a lower efficiency, which increases the fuel consumption and general costs. A higher store of power of the engine is required to accelerate the considerable rotating masses of the hydraulic drive when starting. If the power store is not large the acceleration at the start will prove insufficient and impair the vehicle pick-up speed. In view of these shortcomings the hydraulic drive is mainly used in high-class cars and cross-country vehicles.

The trend in suspension design is to increase considerably the number of elastic elements. Thus, for high-capacity trucks and buses, wide use is made of pneumatic suspensions which keep the floor height constant and reduce weight. Most often, the pneumatic elements of the suspension are applied in combination with leaf springs which act also as guiding devices.

European cars have split rear axles. Independent front wheel suspensions are also fitted to trucks and buses. In addition to reducing unsprung masses, the engine can be positioned lower.

Tyres today are made with the idea in mind of increasing their elasticity and longevity and also reducing their mass. Tyres with a radial arrangement of the cord threads and tubeless tyres with cord made of viscose or nylon become more and more popular.

The speed increase of modern motor vehicles imposes high demands on the braking systems. The shortcoming of drum-type brakes is the appreciable temperature increase of the friction linings during prolonged braking due to inadequate cooling, which tends to reduce the coefficient of friction. For this reason the vehicles often carry disk brakes which are cooled better despite a smaller friction surface. Boosters are applied to obtain high braking moments. In view of their fast response and low mass of brakes with hydraulic drive, it is expedient to use them on trucks of a low, medium or even high load-carrying capacity. In the latter case, vacuum or pneumatic boosters are employed. Towing vehicles utilize pneumatic-hydraulic brakes. With this drive, the brake pedal actuates an air cock whose opened valve admits compressed air which moves the main braking cylinder piston.

The braking dynamic characteristics can also be improved by the use of hydraulic and electric baffles installed behind the gearbox or inside its casing. To make a better use of the wheel grip coefficient special devices are developed and tested which are then incorporated in the brake system to prevent the wheels from being locked. These devices are linked with the wheel and reduce pressure in the brake system when the wheel is locked.

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